



Addis Ababa University
Addis Ababa Institute of Technology
(AAiT)
School of Mechanical & Industrial Engineering
Railway Engineering Stream

**Effect of Track Stiffness Variation on the Dynamic
Response of Passenger Vehicle**

By

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A thesis submitted to the school of Graduate Studies of Addis Ababa
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Declaration

I hereby declare that the work which is being presented in this thesis entitled “**Effect of Track Stiffness Variation on the Dynamic Response of Passenger Vehicle**” is original work of my own, has not been presented for a degree of any other university and all the resource materials used for this thesis have been duly acknowledged.

Theodros Yohannes

Date

This is to certify that the above declaration made by the candidate is correct to the best of my knowledge.

Mr. Tsegaye F. (Advisor)

Date

Acknowledgment

This research is not an output of a single individual. It was made possible with the help of many organizations and people. I wish to express my gratitude to all those who helped me in the completion of this research.

I am very indebted to my advisor, Mr. Tsegaye Feleke for whom I bear high respect and appreciation for his indispensable constructive ideas, sharp critic, insight, guidance and patience. For me the study could not have been accomplished.

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Last but not least, I express my deep appreciation and gratitude to the staffs of Mechanical Engineering Department and Ethiopian Railways Corporation for their kindly support in the course of the study.

Abstract

This paper investigates the effect of track stiffness variation on the dynamic behavior of a railway passenger vehicle. For the analysis, MBS model of the vehicle and the track was developed using SIMPACK software. Two models are developed, one with constant track stiffness and the other with variable track stiffness. The simulation was done on a straight track with no irregularity of rails and wheels. Track stiffness variation may have different causes, however in this research track stiffness variation caused by sleeper spacing was considered.

The dynamic characteristic of railway vehicle greatly depends on the motion of the vehicle. To consider this, the models are allowed to move in three different velocity profiles i.e. the motion of the vehicle during service braking, emergency braking and taking greater acceleration and deceleration values.

For all velocity profiles the outputs of the simulations showed, the vertical and lateral accelerations and displacements of the car body are greater during variable track stiffness case. However, the simulation results for the bogie frame showed an increase in some cases and decrease in the other. From these outputs we can conclude that, track stiffness variation increases the vertical as well as the lateral displacement and acceleration of the car body and it may not have effect on the bogie frame.

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List of Acronyms

AAiT	<i>Addis Ababa Institute of Technology</i>	I_f	<i>Moment of inertia of frame in y-axis</i>
ERC	<i>Ethiopian Railway Corporation</i>	Z_{FFR}	<i>Vertical displacement of front bogie right frame</i>
SIMPACK	<i>Simulation package</i>	Z_{FFL}	<i>Vertical displacement of front bogie left frame</i>
MBS	<i>multi-body system</i>	Z_{BFR}	<i>Vertical displacement of rear bogie right frame</i>
DOF	<i>Degree of Freedom</i>	Z_{BFL}	<i>Vertical displacement of rear bogie left frame</i>
s_0	<i>Position in the track from where the variation is active</i>	α_{FFR}	<i>Front bogie right frame angular motion</i>
C_{yA}	<i>Spring coefficient amplitude in y</i>	α_{FFL}	<i>Front bogie left frame angular motion</i>
C_{vA}	<i>Spring coefficient amplitude in z</i>	α_{RFR}	<i>Rear bogie right frame angular motion</i>
C_{y0}	<i>Constant spring coefficient in y</i>	α_{RFL}	<i>Rear bogie left frame angular motion</i>
C_{z0}	<i>Constant spring coefficient in z</i>	M_w	<i>Mass of wheelset</i>
F_{y0}	<i>Nominal force in y</i>	I_w	<i>Moment of inertia of wheelset in x-axis</i>
F_{z0}	<i>Nominal force in z</i>	Z_{FW1}	<i>Vertical displacement of front bogie front wheelset</i>
d_{y0}	<i>Constant damper coefficient in y</i>	Z_{FW2}	<i>Vertical displacement of front bogie second wheelset</i>
d_{z0}	<i>Constant damper coefficient in z</i>	Z_{RW1}	<i>Vertical displacement of rear bogie front wheelset</i>
F_y	<i>Lateral track force</i>	Z_{RW2}	<i>Vertical displacement of rear bogie second wheelset</i>
F_z	<i>Vertical track force</i>	w_{FW1}	<i>Front bogie first Wheelset angular motion</i>
r_{rely}	<i>Marker distance in y</i>	w_{FW2}	<i>Front bogie second Wheelset angular motion</i>
r_{relz}	<i>Marker distance in z</i>	w_{RW1}	<i>Rear bogie first Wheelset angular motion</i>
l_{nomz}	<i>Nominal length in vertical direction</i>	w_{RW2}	<i>Rear bogie second Wheelset angular motion</i>
l_{y0}	<i>Unstretched length y ($s = s_0$)</i>	ρ_s	<i>Secondary suspension damping</i>
l_{z0}	<i>Unstretched length z ($s = s_0$)</i>	k_s	<i>Secondary suspension stiffness</i>
V_{rely}	<i>Velocity in y</i>	ρ_p	<i>Primary suspension damping</i>
V_{relz}	<i>Velocity in z</i>	k_p	<i>Primary suspension stiffness</i>
n_z	<i>Sinus exponent in z</i>	ρ_t	<i>Track damping</i>
n_y	<i>Sinus exponent in y</i>	k_{TFR1}	<i>Front bogie first wheelset right track stiffness</i>
f_s	<i>Sleeper frequency</i>	k_{TFL1}	<i>Front bogie first wheelset left track stiffness</i>
M_c	<i>Mass of car body</i>	k_{TFR2}	<i>Front bogie second wheelset right track stiffness</i>
I_{cx}	<i>Moment of inertia of car body in x axis</i>	k_{TFL2}	<i>Front bogie second wheelset left track stiffness</i>
I_{cy}	<i>Moment of inertia of car body in y axis</i>	k_{TRR1}	<i>Rear bogie first wheelset right track stiffness</i>
Z_c	<i>Motion of car body vertically</i>	k_{TRL1}	<i>Rear bogie first wheelset left track stiffness</i>
θ_c	<i>Angular car body displacement about x axis</i>	k_{TRR2}	<i>Rear bogie second wheelset right track stiffness</i>
α_c	<i>Angular car body displacement about x axis</i>	k_{TRL2}	<i>Rear bogie second wheelset left track stiffness</i>
l	<i>Half length of the car body</i>		
j	<i>Half the wheel base</i>		
b	<i>Half the distance between wheel-rail contacts</i>		
M_f	<i>Mass of frame</i>		

CHAPTER ONE

INTRODUCTION

1.1 Background

The history of railways dates back nearly 500 years, and includes systems with man or horse power and rails of wood or stone. Modern rail transport systems first appeared in England in the 1820s. These systems, which made use of the steam locomotive, were the first practical forms of mechanized land transport, and they remained the primary form of mechanized land transport for the next 100 years. In January 1888, Richmond, Virginia served as a proving ground for electric railways. By the 1890s, electric power became practical and more widespread, allowing extensive underground railways. [24]

The railway line construction in Ethiopia was first started in October 1897 from Djibouti in the period of Emperor Menelek II. The first commercial service began in July 1901, from Djibouti to Dire Dawa. By 1915 the line reached Akaki and two years later came all the way to Addis Ababa. Now, rail transport in Ethiopia consists only of trial services from Djibouti to Dire Dawa on a reconstructed line. In January 2010, it was announced that the Ethiopian government had signed a memorandum of understanding with four companies to build a new rail network. The network will be 5,000 km long, and radiate from Addis Ababa. [24]

The experience of operating lines with high-speed services in Europe over the last 20 years (1981- 2001) has shown that the railway can achieve market shares of up to 50 percent on any given intercity route, taking into consideration all the other competing modes of transport (airplane, private vehicle and coach), or 80 and even 90 percent with respect to air transport alone. From the commercial point of view, this reality has resulted in a significant increase in the income obtained by the railway, which has been accompanied by limited operating costs. [5]

It is therefore reasonable to assume that, besides the effort that the railway is making to ensure that its commercial offer is more attractive and competitive, it is also advisable to try to design infrastructures that require as little maintenance as possible. This obvious assumption acquires greater practical interest if it is borne in mind that the maximum commercial service running speed is expected to increase over the next few years.

The railway train running along a track is one of the most complicated dynamical systems in engineering. Many bodies comprise the system and so it has many degrees of freedom. The bodies that make up the vehicle can be connected in various ways and a moving interface connects the vehicle with the track. This interface involves the complex geometry of the wheel tread and the rail head and nonconservative frictional forces generated by relative motion in the contact area. [4]

The extrapolation of conventional railway technology to higher speeds has led in many cases to increased traction forces, increased wheelset mass, and greater track stiffness. New problems of interaction between vehicle and track have emerged, such as track irregularities, ballast settlement and deterioration, increased levels of rail corrugation, out-of-round wheels, and noise. The solution of these problems requires the consideration of structural dynamics of both vehicle and track.

Track irregularities cause oscillation or vibration of the train, and this may induce discomfort for passengers and damage for parts of the vehicle and track and the goods that are being transported. Among the track irregularities, track stiffness variation along the track is the major one. Track stiffness variations may be more difficult to deal with because even a track with an ideal track geometry may hide irregularities that are not discovered until the track is loaded by the train. [3] The wheel will be influenced by the varying stiffness of the track. As the track is stiffer at the sleepers, and softer in between. Other places along the track having variable stiffness are at switches, turnouts and transition places. [2]

In this paper, multi-body simulation (MBS) tool is used to model a passenger car including the track flexibility. Two cases (models) are developed to analyze the influence of track stiffness variation on the dynamic behavior of vehicle components. One model is developed with constant track stiffness and the other with variable stiffness. The dynamic loading of the vehicle components will be analyzed for both case and the outputs will be compared. The results of the analysis will be an input for different disciplines and studies.

1.2 Significance of the Research

The variations of track stiffness will induce variations in the wheel-rail contact force. This intensifies track degradation rate and increases the vibration of the components of the vehicle. As soon as the track geometry starts to deteriorate the variation of track stiffness increases. This increases the wheel-rail interaction forces. In addition, the dynamic component of wheel-rail contact force, as generated by track stiffness variation, is an important source to ground vibration and ground-borne noise.

In this research the influence of track stiffness variation on the vehicle dynamic system will be investigated. Understanding the effect of track stiffness variation on the dynamic behavior of the vehicle has paramount value for different sections of railway companies. From track maintenance point of view, the dynamic loading determines a track section's maintenance requirement. By understanding vehicle-track dynamic loading on track stiffness variation points, maintenance engineer would be able to control the cause of the dynamic loading and thus be able to optimize his maintenance input. During the design of vehicle components the dynamic loading should have to be considered. So, this research gives clear insight to design engineers about the dynamic loading caused by track stiffness variation. This research will also be an input for researches that focus on the study of passenger comfort and noise and vibration control.

1.3 Expected output of the research

As it has been discussed, understanding the effect of track stiffness variation can be used as input to different studies and activities. Considering this the following outputs are expected from this research:-

- Fully developed multi-body dynamic model of the vehicle including flexible track using SIMPACK software. The developed model can be used for further researches with little modification.
- Mathematical model of vehicle-track system and the governing equation of motion.
- Vertical and lateral displacements of the car body and the bogie frame.
- Vertical and lateral accelerations of the car body and the bogie frame.

1.4 Objective of the research

1.4.1 General objective

- The general objective of the research is to study the influence of track stiffness variation on the dynamic response of railway vehicle.

1.4.2 Specific objective

- Discuss different researches that have been done in relation to the research.
- Develop the mathematical model of the vehicle-track system and derive the governing equations of motion.
- Develop the model in a multi-body simulation tool (SIMPACK).
- Calculate the vertical as well as the lateral displacements and accelerations of the vehicle components.

1.5 Research Methodology

The research begins with conceptual literature study and internet surfing to develop and structure suitable data collecting mechanism. The literature also will be reviewed side by side with information acquired from the primary as well as secondary data. The general methodology followed throughout the research is depicted in the following schematic diagram.

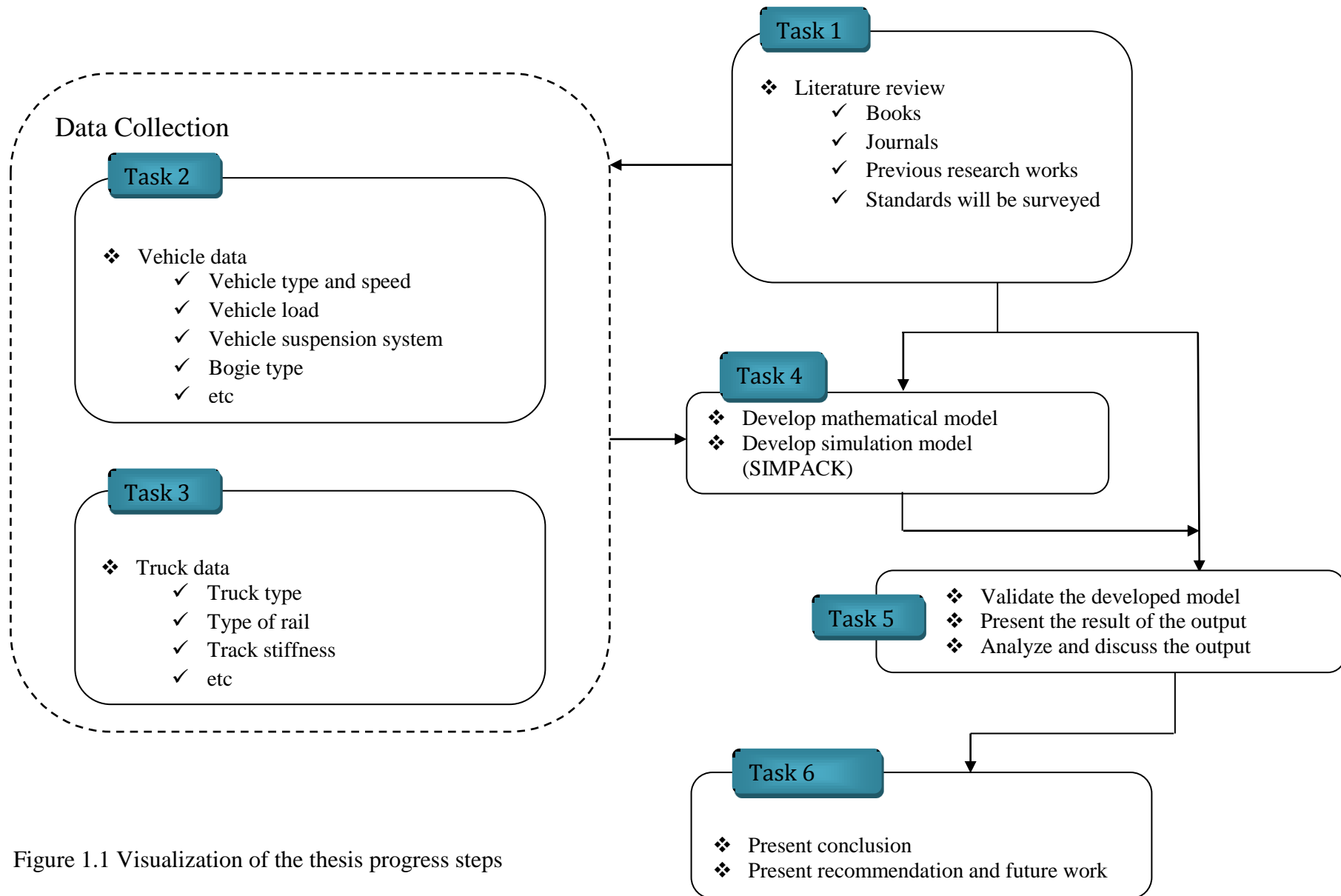


Figure 1.1 Visualization of the thesis progress steps

1.6 Organization of the Research

The research is organized in five main chapters and each chapter comprises subtopics. Chapter one covers the necessary background to the study. In chapter two, the review of some of the journal articles, proceedings and publications which were referred during the course of the research are presented and the need to do this research was delineated. The third chapter covers the modeling issues. In this chapter, the physical, mathematical and MBS model of the passenger vehicle are developed. In addition, the governing equation of motion was derived. In chapter four, the developed model was validated and the outputs of the SIMPACK model was analyzed and discussed. The fifth chapter is dedicated for conclusion, recommendation and future work.

1.7 Scope, Limitation and Assumption of the Research

1.7.1 Scope of the research

- The variation of track stiffness can be occurred at different places of the track such as switches, transition zones, hanging sleepers, culverts, etc. However, in this research only track stiffness variation occurred due to sleeper spacing will be studied.
- The dynamic characteristic of the vehicle includes numerous types of analysis such as modal and frequency analysis. However, in this research the displacements and accelerations of the vehicle components will be studied. In addition the vehicle components include number of parts; here the analysis will be done for the car body and bogie frame.

1.7.2 Limitation of the Research

- Modeling and analyzing the dynamic characteristics of vehicle-track system needs high capacity computers. Results obtained on this thesis might have been refined in the presence of this resource.
- Real vehicle and track data such as suspension system stiffness and damping values, dimension of bogie components, track stiffness and damping values could not be accessed from the concerned organizations. So, data was traced and collected from similar researches and journals.

1.7.3 Assumptions

- All the components of the vehicle are modeled rigid.
- The dynamic simulation was made on straight track with no superelevation and curve radius.
- All wheel profiles were considered identical from left to right on a given axle and from axle to axle and perfect wheels with no defects are considered.
- During simulation an ideal velocity profile was used.
- Non linearity of the suspension system components was neglected and taken as linear systems.
- No rail head defect was considered such as corrugation.

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction

Track quality plays an important role in railway maintenance cost, noise level, and vehicle safety. One of the affecting factors of track quality is the track stiffness. Track stiffness is defined as ratio of the load applied to the rail to vertical rail deflection. Track modulus, on the other hand, is a measure of the vertical stiffness of the track foundation, [3]. In technical texts, track stiffness is presented by K and is measured in N/mm while modulus of the rail support is indicated by u and is measured in Pa., [8].

Stiffness variation on the track can be induced by different causes. The wheel will be influenced by the varying stiffness of the track due to the sleeper spacing as the track is stiffer at the sleepers, and softer in-between. The rail manufacturing process also may induce rail irregularities. Other places along the track having variable stiffness are at switches and turnouts. Especially if a switch is equipped with a manganese crossing (frog), the rail bending stiffness (EI) will change dramatically at the crossing. [2]

2.2 Track Stiffness

The vertical stiffness of a railway track plays an important role when considering maintenance work, but it is also an important factor when looking at dissipated energy of a train. [5] Proposed optimal values of the vertical stiffness. They optimized the maintenance costs and the costs for dissipated energy of a train versus the vertical track stiffness. Their result shows that the optimum vertical track stiffness should be between 70 and 80 kN/mm.

[6] proposed a method to predict track deterioration due to dynamic wheel loading and spatially varying track stiffness. Findings from the research show the significant influence of spatial track stiffness variations on track deterioration, both in terms of the dynamic loading of track components and differential track settlement.

[3] discussed the possibility to smooth out track stiffness variations and demonstrated that by modifying the stiffness variations along the track, for example by use of grouting or under-sleeper pads, the variations of the wheel/rail contact force may be considerably reduced.

[14] analyzed the influence of the track stiffness and damping parameters on wheel out-of-roundness and showed the higher the vertical track stiffness the quicker the out-of-round profiles grow. That means out-of-round wheels will occur often in modern high-speed passenger trains because the new high-speed tracks are stiffer than the older ones.

Taking three different train models, [15] presented an efficient method for analyzing an influence of random changes in ballast stiffness on rail way bridge vibration caused by a train passage. The result demonstrates that a simplified model of a train may be used in stochastic displacement analysis. In turn, the acceleration analysis requires more realistic model of the vehicle. Random fluctuation of the ballast stiffness has a much greater impact on accelerations of bridge vibration than on displacement.

[16] analyses the track stiffness from the aspect of its influence on the quality of the vertical track geometry. The paper analyses optimum stiffness. Optimum stiffness is conditioned by the single stiffness of all the elements of the superstructure as well as by their mutual compatibility.

2.3 Track Modulus (stiffness) Measurement

The availability of systems that accurately measure and monitor vertical track (and rail) irregularities facilitates maintenance management. [10] surveyed such systems, covering methods measuring unloaded or loaded track irregularities in the wavelength interval relevant for the excitation of ground-borne vibration and noise. The main features of the different systems are listed and *pros* and *cons* are discussed.

[8] described the design of a system for on-board, real-time, non-contact measurement of track modulus (track stiffness). The proposed system is based on measurements of the relative displacement between the track and the wheel/rail contact point. A laser-based vision system is used to measure this relative displacement. Then a mathematical model is used to estimate track modulus.

Sandy desert areas are critical regions about the contamination of ballast. In these areas, flowing sand grains influence between ballast aggregates and increase the stiffness of ballast layer and the rail support modulus. [9] Investigated the results of a field investigation about the effect of ballast contamination on the values of the rail support modulus in sandy desert areas.

[11] proposed a system to make real-time measurements of the vertical track modulus from a moving railcar. They explained the basic concept and presented some field tests. The displacement and modulus data obtained suggest such information could be used to better plan track maintenance cycles.

[12] have studied sensitive wavelength of track irregularity and its influence on dynamic responses of train-bridge system in high-speed railway and concluded that the sensitive wavelength of track irregularity on bridge is affected by bridge characteristics, vehicle model and moving speed.

[13] have done a research on measurements and analysis of track irregularities on high speed maglev lines and they proposed a measurement principle and processing method and showed that the track quality of the maglev line is better than that of a railway line when the wavelength of the irregularities is larger.

2.4 Summary of Literature Review

As it can be seen from the surveyed literatures, most researchers focused on the study of the effect of track stiffness variation on the track itself. They have studied where track stiffness variation occurs and how this variation can be optimized. The major focus of the researchers was to reduce maintenance cost of the track which is seen to be higher compared to the maintenance cost of the train. Other researchers focused on the developing of the methods of measuring track stiffness (modulus) along the track. The results of the measurement can be used to determine the sections of the track that has pronounced deterioration. However, the causes of the track deterioration cannot be determined from such measurements. In this research, the effect of track stiffness variation on the dynamic behavior of the vehicle components will be studied. This has got less attention by the researchers.

CHAPTER THREE

MODEL OF VEHICLE/TRACK SYSTEM DYNAMICS

3.1 Physical Passenger Car Model

The model used for the analysis is passenger car which is going to be supplied to Ethiopian Railways Corporation. The maximum operation speed of the car is 120 km/h. The car has two bogies at the front and back. The bogie type is a three piece bogie (209P) with a construction speed of 140km/h. [18] Figure 3.1 shows similar car body. The technical specification of the vehicle is also given in Table 3.1 [18]

Table 3.1 Main technical parameters of passenger Vehicle (car).

Parameter	Value
Design speed	140km/h
Operating speed	120km/h
Model No	YZ25B
Service life	30 Years
Max. configuration	20 car per train set
Min. curve negotiable for the train	145m
Ride index	$W \leq 2.5$
Noise (120km/h)	$\leq 68\text{dB(A)}$
Gauge	1435mm (standard gauge)
Fixed passenger capacity per car	118 persons
Car body length	25500mm
Car body width	3105mm
Vehicle center distance	18000mm
Height from rail top to car floor surface	1283mm
Car body centerplate to rail top	780mm
Bogie type	209P
Axle base	24000mm
Wheel diameter	915mm
Axle load	$\leq 17\text{t}$



Figure 3.1 Physical model of the car going to be analyzed. [19]

3.2 Mathematical Model of Track Dynamics

Depending on what one wants to investigate, railway track structure components may be modelled in a simpler or a more sophisticated manner. In this section the present models used in vehicle/track dynamic analysis will be presented and model used in this research will be described in detail.

3.2.1 The present models used

3.2.1.1 Beam (Rail) on Continuous Elastic Foundation (Winkler Beam)

In the most simple track model, a beam (that is a model of the rail) rests on a continuous elastic foundation. The foundation is modeled by evenly distributed linear spring stiffness. The distributed force supporting the beam is then proportional to the beam deflection. (Figure 3.2)

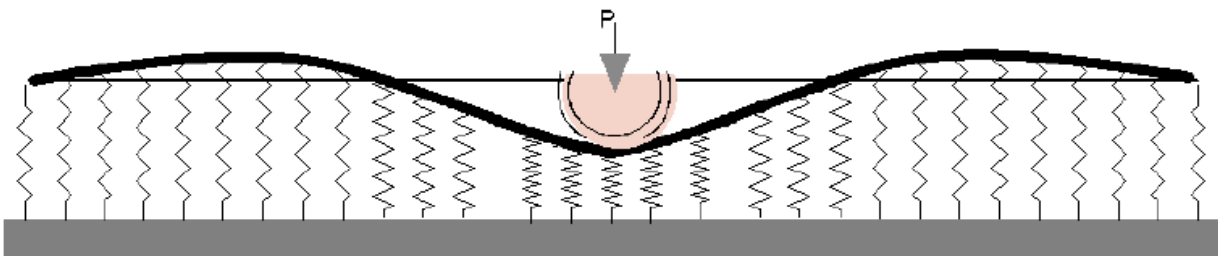


Figure 3.2 Beam on elastic foundation loaded with a point force P from the wheel. The thick line indicates beam (rail) deflection due to the wheel load P. [7]

3.2.1.2 Vehicle – Bridge Interaction (Moving Mass on Simply Supported Beam)

In the previous track model, only the railway track has been considered a dynamic system. A railway bridge is also a dynamic system that interacts with a train passing the bridge. The dynamic interaction of a wheel (a rigid mass) and a simple bridge model (a simply supported beam) is now being investigated. (Figure 3.3)

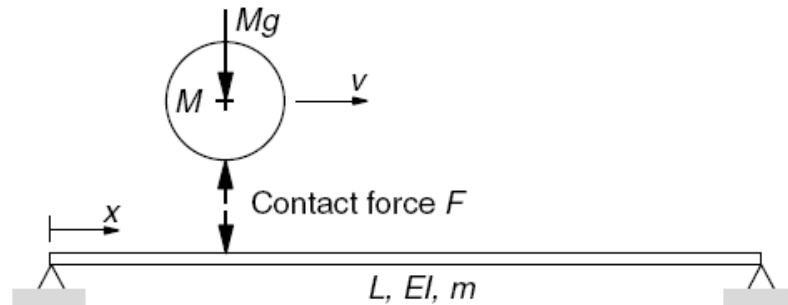


Figure 3.3 Vehicle - bridge interaction model of railway track. [2]

3.2.1.3 Beam (Rail) on Discrete Supports

Rail is modelled as a beam (Euler – Bernoulli or Rayleigh – Timoshenko beam theory), the railpads are modelled by spring – damper systems, the sleepers are rigid masses, and the ballast is modelled by spring – damper systems. (Figure 3.4)

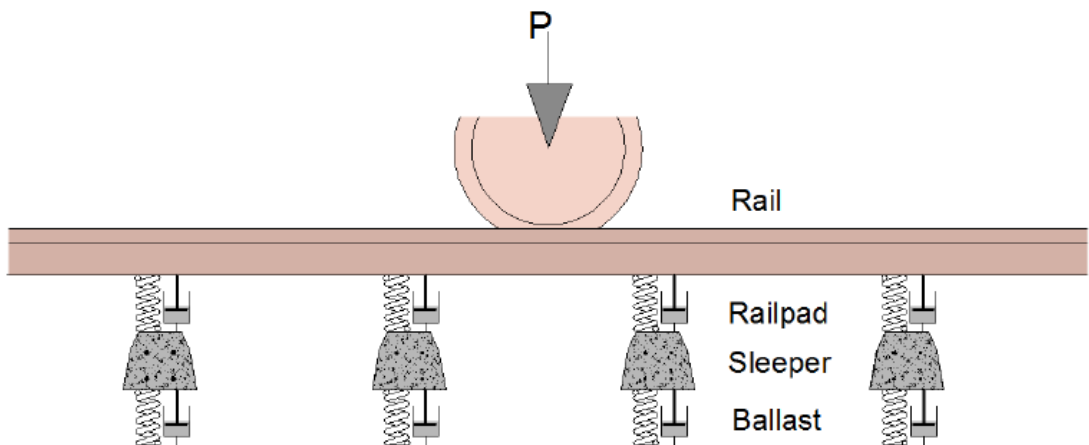


Figure 3.4 Rail on discrete supports. [7]

3.2.1.4 Discretely Supported Track Including Ballast Mass

By this model, the four resonance vibration modes (a) embankment vibration, (b) track-on-the-ballast vibration, (c) rail-on-rail pad vibration, and (d) pinned – pinned vibration of the rail, may be captured. (Figure 3.5)

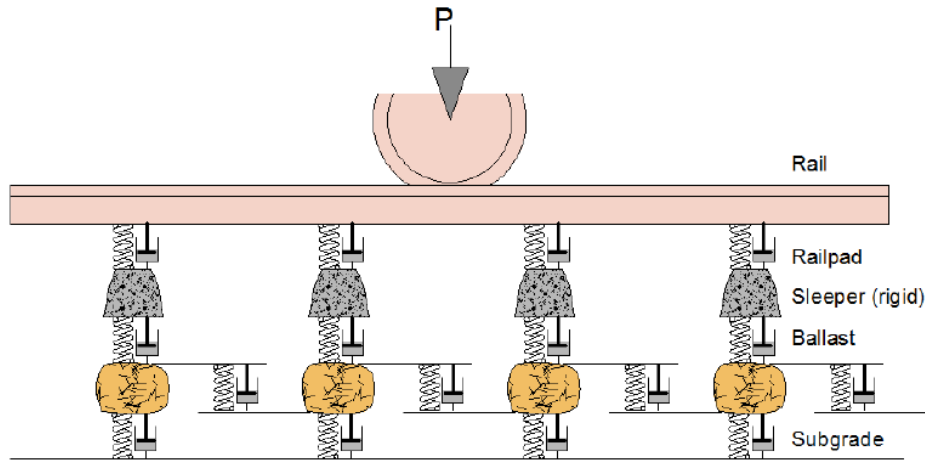


Figure 3.5 Rail on discrete supports with rigid masses modeling the sleepers. [7]

3.2.1.5 Rails on Sleepers Embedded in Continuum. Three-Dimensional Finite Element Models

The most realistic track model, and the model that normally requires the most computer capacity, is the model where rails and sleepers are modeled as beams (or possibly as three-dimensional bodies) with elastic elements modeling the rail pads between the rails and the sleepers. The sleepers are embedded in a continuous medium. This requires that the track bed is modeled by three dimensional finite elements. (Figure 3.6)

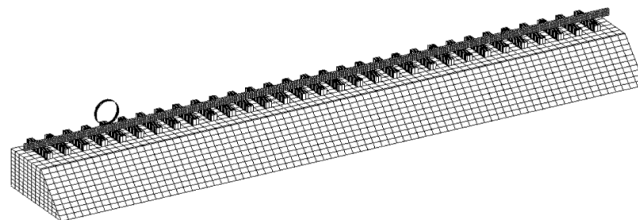


Figure 3.6 Rails on Sleepers Embedded in Continuum, Three-Dimensional Finite Element Model. [2]

3.2.2 Model used in the research

In this research a simplified discrete track support model will be used because the study focuses on the vehicle dynamic performance. If the track behavior was the major objective a detailed model of the track is necessary. As it is known, the track is stiffer at the sleepers and less in between. This variation of stiffness should have to be included in the model. In MBS (SIMPACK) software this can be done in different ways. Usually, a track model with moving equivalent mass is used in multi-body simulations. In this study, this modeling technique will be used. The moving equivalent mass represents the sleeper and a part of the track oscillating together with the wheelset. This simple model of moving elastic track contains stiffness and damping in the vertical and lateral directions. Default values for the stiffness and damping are given for ballasted track [22] (Table 3.2). The stiffness and damping variation caused by sleeper spacing can be represented by sinusoidal curve with the variation being the amplitude.

Table 3.2 Stiffness and damping values of the track.[22]

Parameter	Value
Lateral stiffness (per rail)	20 MN/m
Vertical stiffness (per rail)	75 MN/m
Lateral damping (per rail)	0.6 MNs/m
Vertical damping (per rail)	1.5 MNs/m
Lateral stiffness amplitude (per rail) C_{yA}	3210 N/m
Vertical stiffness amplitude (per rail) C_{vA}	6340 N/m

Force law for operating mode "Sinusoidal"

Stiffness and damping vary in a sinusoidal way after the start position s_0 in track is reached.

For $s(t) < s_0$:

$$F_y = C_{y0}(r_{rely} + l_{y0}) + d_{y0}V_{rely} \quad 3.1$$

$$F_z = C_{z0}(r_{relz} - l_{nomz} + l_{z0}) + d_{z0}V_{relz} \quad 3.2$$

With l_{y0} and l_{z0} determined from the nominal forces and the initial stiffnesses:

$$l_{y0} = \frac{F_{y0}}{C_{y0}} \quad \text{and} \quad l_{z0} = \frac{F_{z0}}{C_{z0}} \quad 3.3$$

For $s(t) \geq s_0$:

$$F_y = C_y(s)(r_{rely} + l_{y0}) + d_y(s)V_{rely} \quad 3.4$$

$$F_z = C_z(s)(r_{relz} - l_{nomz} + l_{z0}) + d_z(s)V_{relz} \quad 3.5$$

With l_{y0} and l_{z0} the same as above

$$C_y(s) = C_{y0} + C_{yA} \sin^{n_y}(f_s(s(t) - s_0)) \quad 3.6$$

$$C_z(s) = C_{z0} + C_{zA} \sin^{n_z}(f_s(s(t) - s_0)) \quad 3.7$$

During nominal forces calculation:

$$F_y = C_{y0}r_{rely} + d_{y0}V_{rely} + F_{y0} \quad 3.8$$

$$F_z = C_{z0}(r_{relz} - l_{nomz}) + d_{z0}V_{relz} + F_{z0} \quad 3.9$$

With F_{y0} and F_{z0} as nominal forces, in order to make the nominal forces calculation work.

3.3 Mathematical Model of Vehicle Dynamics

The considered vehicle is a passenger car which is equipped with a pair of two bogies with double suspension systems. The wheelset and the bogie are connected by the primary suspension, while the car body is supported on the bogie through the secondary suspension. All the components (car body, bolsters, side frames, wheelsets) modeled as rigid body. The mathematical model of the vehicle is given in Figure 3.8 and the equation of motion is described in detail.

3.3.1 Rail Vehicle/Track Modeling Axis System Used In the Research

A rail vehicle model is a collection of inertial elements (masses and wheelsets) connected by suspension elements and/or constraint elements. The model requires an associated axis system as shown in Figure 3.7 positions of inertial elements and suspension elements are specified in terms of longitudinal position, lateral position and height above rail. The longitudinal axis (X) is

fixed at the mid-point of the vehicle or train; positive longitudinal positions are in the direction of travel. The lateral axis (Y) is relative to the track centreline. The vertical axis (Z) is relative to the height above the rail which is considered positive downward. The same axis system will be used in SIMPACK software packages.

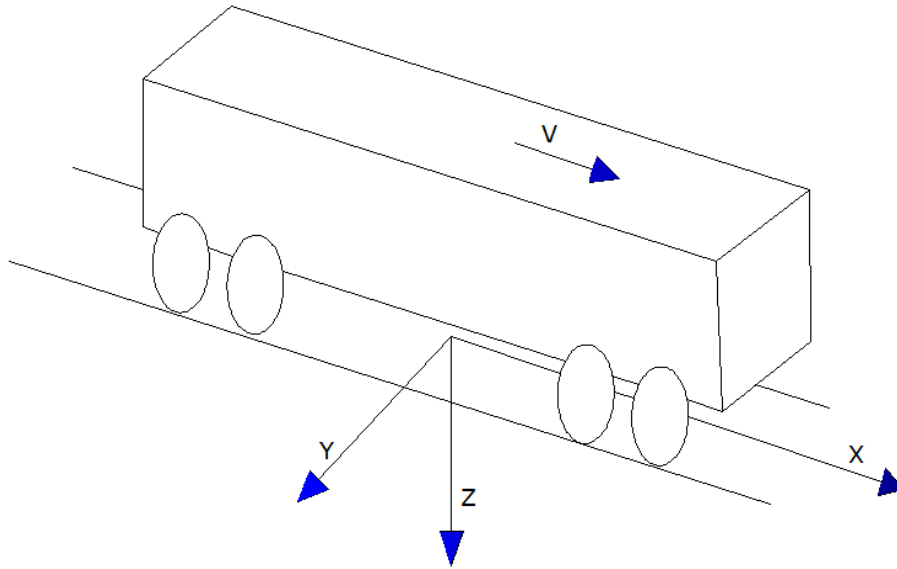
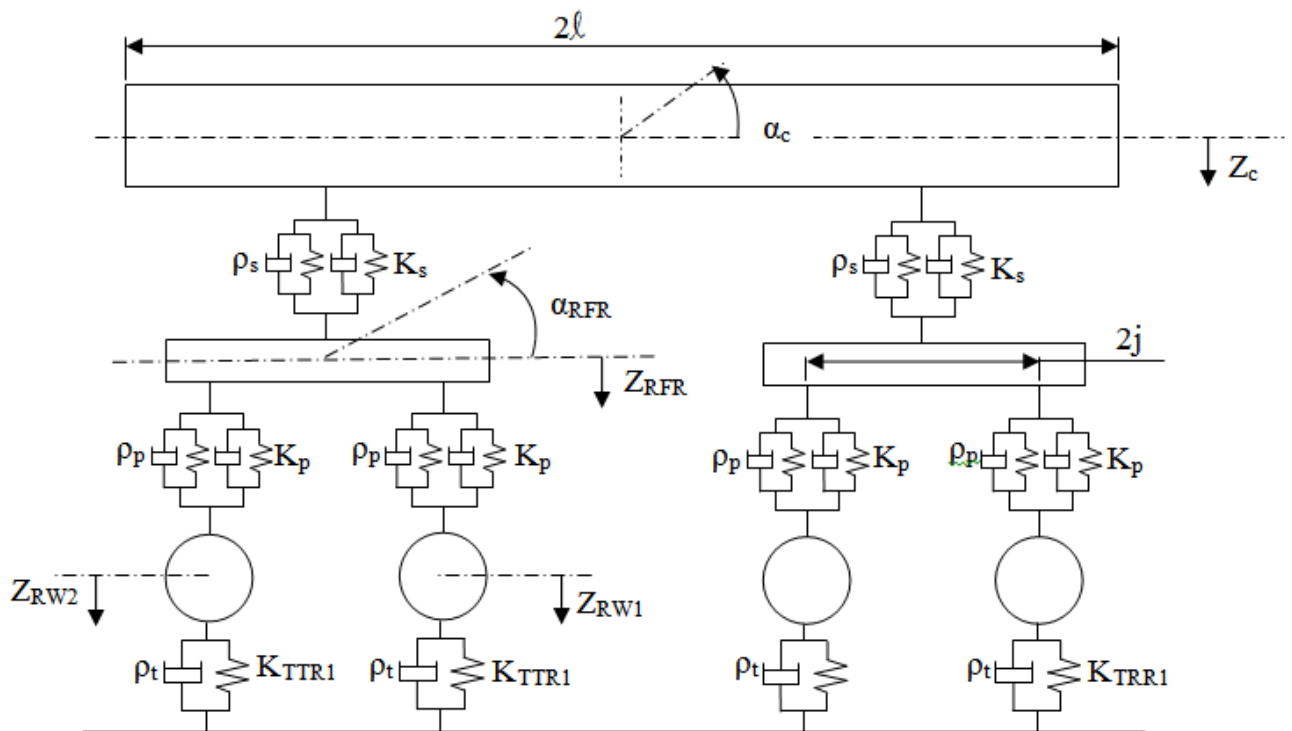


Figure 3.7 Vehicle modeling axis system used in the research.



(a)

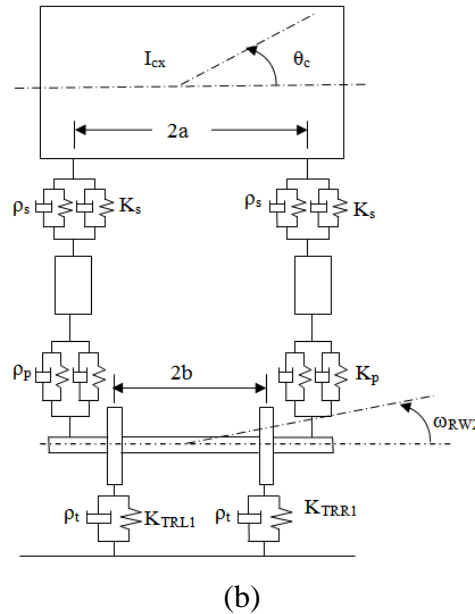


Figure 3.8 Coupled vehicle/track model. (a) Side view (b) Rear view

3.4 MBS Model Development Using SIMPACK

Multi-body simulation, known also as multi-body system (MBS) simulation, is used to predict and optimize the behavior of any type of multi-body system by solving the equations of motion. The name multi-body can refer to a wide range of systems including machinery, vehicles, robotics, biomechanics, etc. The bodies of a multi-body system are linked by means of joints which allow certain relative motions and restrict others. The bodies themselves can be rigid or flexible. In describing the kinematic behavior, the motion or the position of the multi-body system is studied with respect to the kinematic joints. Dynamic problems describe the motion of the system due to the applied forces and the inertia characteristics of the bodies, i.e. their mass, moments of inertia and position of the center of gravity. [21] Following a brief description of the model development process in SIMPACK Release 9.3 will be presented.

3.4.1 Developing guided wheelset

For creating the guided wheelset one can use the existing Rail-Wheelset module or start developing from scratch. In this research starting from the scratch method was used with the following procedure. The final guided wheelset model is shown in Figure 3.9.

Set Up the Wheelset Model:- According to our sign convention, the z-axis points downwards and the y-axis points to the right when looking in forward direction as shown in Figure 3.7. To do this follow the following procedure:-

- Create a new SIMPACK model using the 'General' template.
- Switch the gravity direction on the Global parameter from -9.81 to 9.81.
- Rotate the view so that the x-axis points forward, the y-axis to the right and the z-axis downwards.
- Save this view.

Create the wheelset shaft:- Change the available Body to cylinder and give the appropriate parameters and states as given in the Table 3.3. Change the joint type to 7:General rail-track joint. Copy and paste the Body.

Create wheelset carrier body:- Insert new Body and give a mass of 60Kg considering one meter length of the rail. Then change the type to RWC rail. Copy and paste three times to get the four wheel carrier bodies. Insert Rail-Wheel pair, on the property dialog on the rail tab change the mounting to Body (Moving ballast mass). Open the Body property of RWC rail and give the appropriate Rail-Wheel pair. Change the Joint type of the rail Body to 91:Wheel/Rail Track Sleeper.

Table 3.3 Main parameters required for developing the guided wheelset.

Parameter	Value
Wheelset mass	1000 Kg
Wheelset shaft length	2m
Wheelset shaft diameter	0.16m
Wheel profile type	S1002
Rail profile	UIC60
Rail cant	1:40
Young's modulus	202GN/m ²
Poisson number	.277
Friction coefficient	0.4

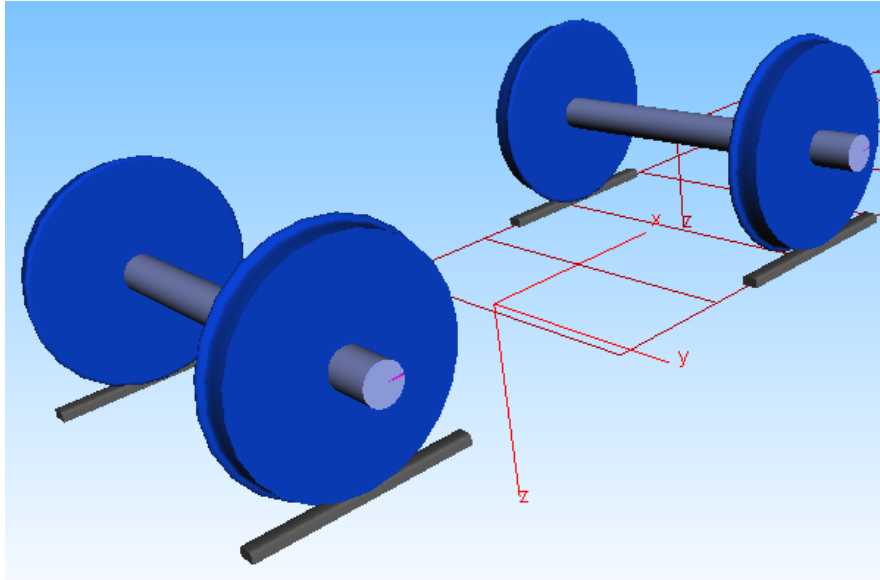


Figure 3.9 The guided wheelset model

3.4.2 Developing Bogie

The developed bogie with included frame and the bolster is shown in Figure 3.11.

Develop the side frames:- Input a Body and change the type to 22: Wheel Rail Bogie. Then modify to make it three piece bogie based on the dimensions given in the Table 3.4. Change the joint type to 7: General rail-track joint.

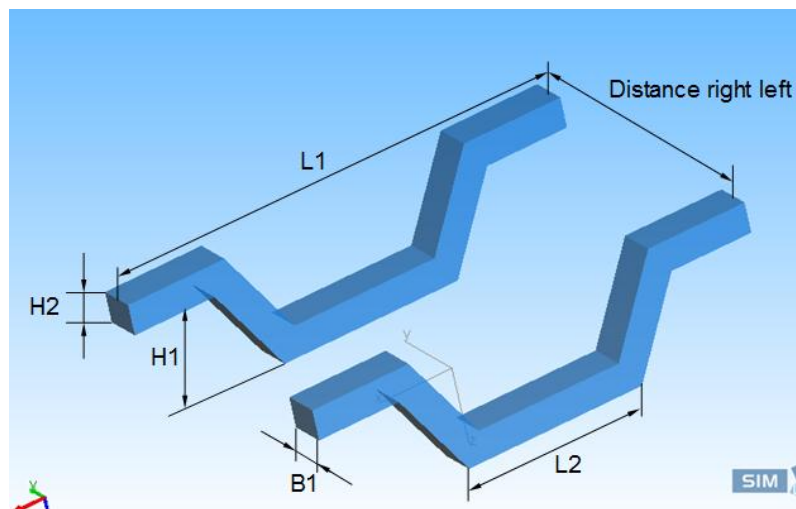


Figure 3.10 Bogie frame dimensions

Table 3.4 Dimensions required for developing the Side frames and the bolster.

Parameter	Value
Bogie frame mass (total)	1140 Kg
L1	3.2m
L2	1.2m
H1	-0.6m
H2	0.2m
B1	0.2m
Distance right left	2m
Bolster mass	220Kg
Bolster dimension (X-Y-Z)	0.7mx2.3mx0.12m

Insert the bolster:- insert new body type 1: Cuboid and give the appropriate dimension as shown in Table 3.4 and create the centre plate. Insert a marker of type 1: Identity Matrix at the centre of the Centre plate. This marker will be used for joining the car body with the bolster.

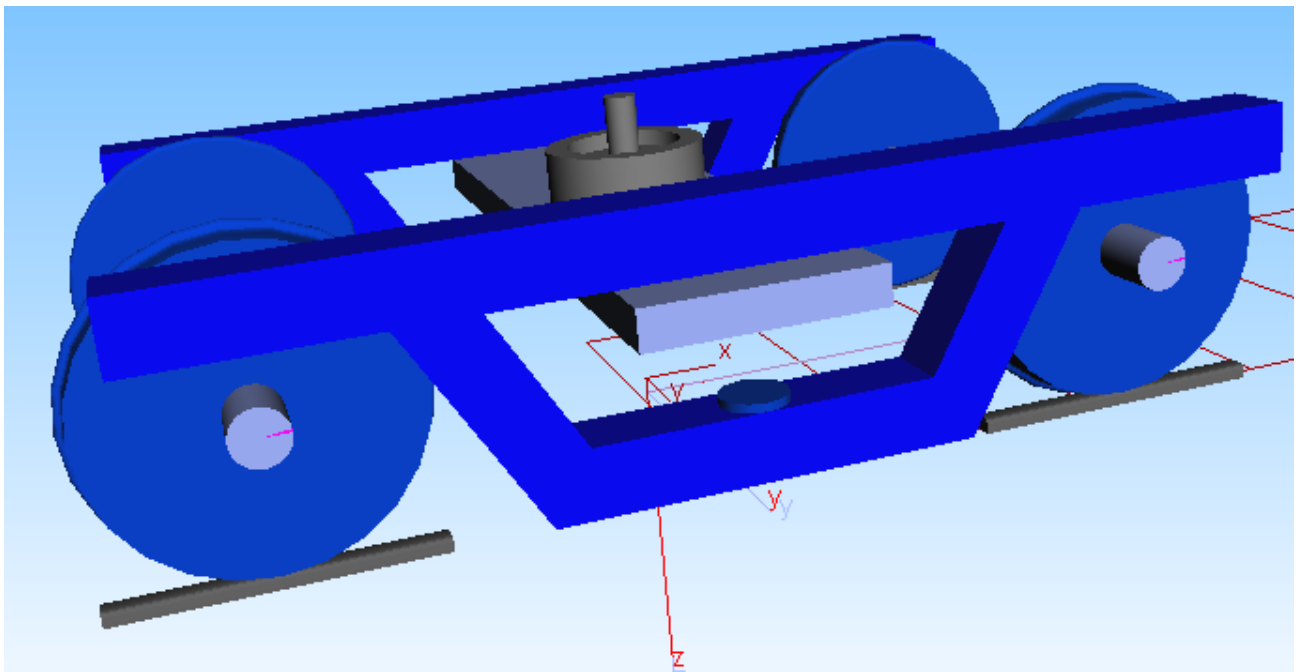


Figure 3.11 Developed bogie frame and bolster

3.4.3 Developing bogie suspension elements

The developed bogie including the suspension elements is shown in Figure 3.12.

Creating markers and force elements: to create the primary spring suspension insert markers type 1: Identity Matrix on the frame and wheelset on the attachment points of the primary suspension. Insert force element type 86: Spring Damper Ser/Par cmp between the two markers. Again insert the same type of markers on the same places. Then insert force element type 6: Spring Damper Serial cmp between the two markers. To create the secondary suspension follow the same step using force types 79: Shear spring cmp and 6: Spring damper Serial ptp. Table 3.5 gives the data required for the development.

Creating the primitives:- insert markers of type 88: Point to Point between the previous markers on the reference system and insert primitive type 13: Spring using this marker. In order to create the damping element perform the same step but use the primitive type 19: Point to Point and add a primitive type 7: Rotational Body.

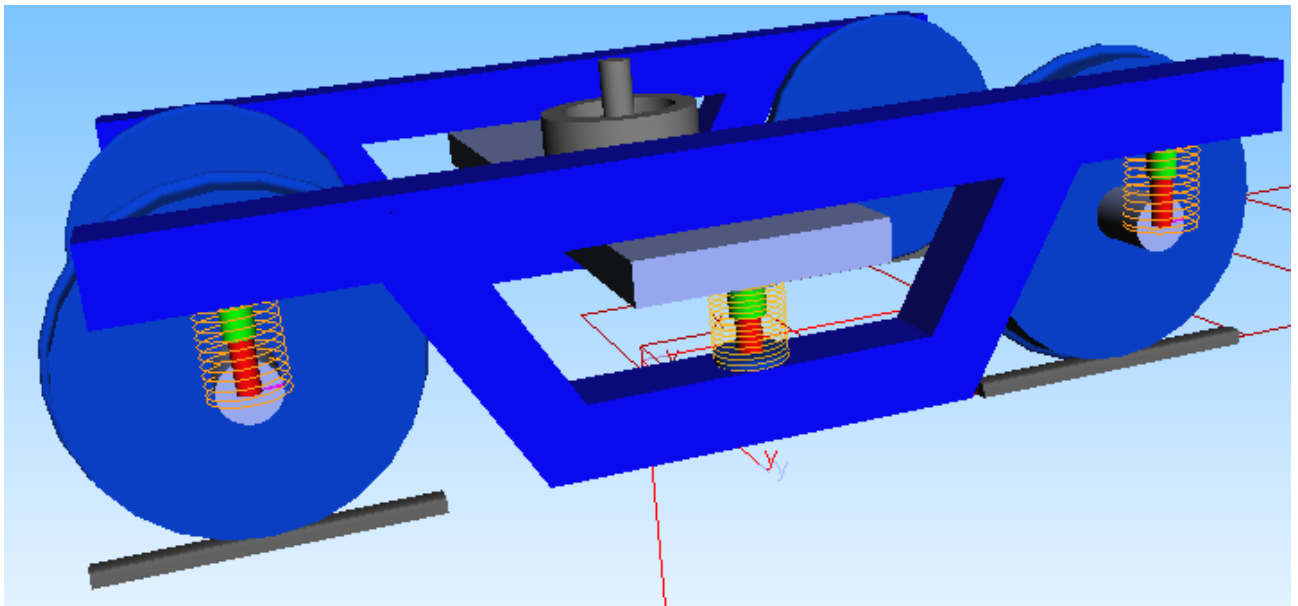
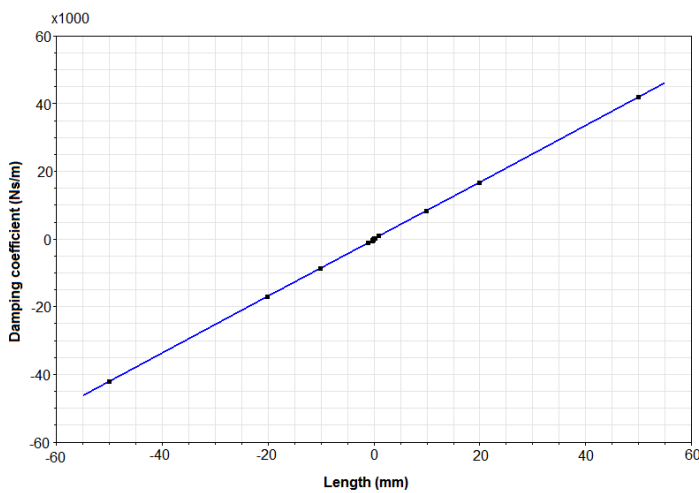


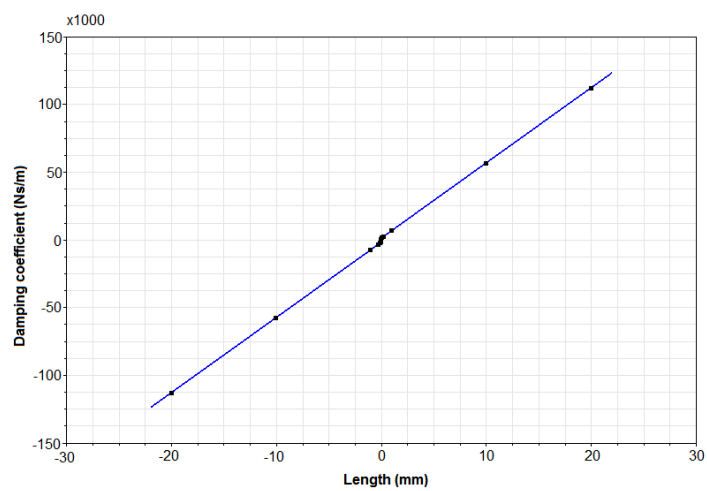
Figure 3.12 Bogie with the suspension elements.

Table 3.5 Stiffness and damping values for the primary and secondary suspension

Parameter	Value
Longitudinal primary suspension spring stiffness	60 MN/m
Lateral primary suspension spring stiffness	7.5 MN/m
Vertical primary suspension spring stiffness	60 MN/m
Longitudinal primary damping coefficient	15000Ns/m
Lateral primary damping coefficient	2000Ns/m
Vertical primary damping coefficient	Variable (Figure 3.13)
Longitudinal secondary suspension spring stiffness	160000 N/m
Lateral secondary suspension spring stiffness	160000 N/m
Vertical secondary suspension spring stiffness	430000 N/m
Lateral secondary damping coefficient	17500 Ns/m
Roll bending stiffness for secondary suspension	10500 N/m
Pith bending stiffness for secondary suspension	10500 N/m
Torsional bending stiffness for secondary suspension	10500 N/m
Secondary damping coefficient	Variable (Figure 3.13)



(a)



(b)

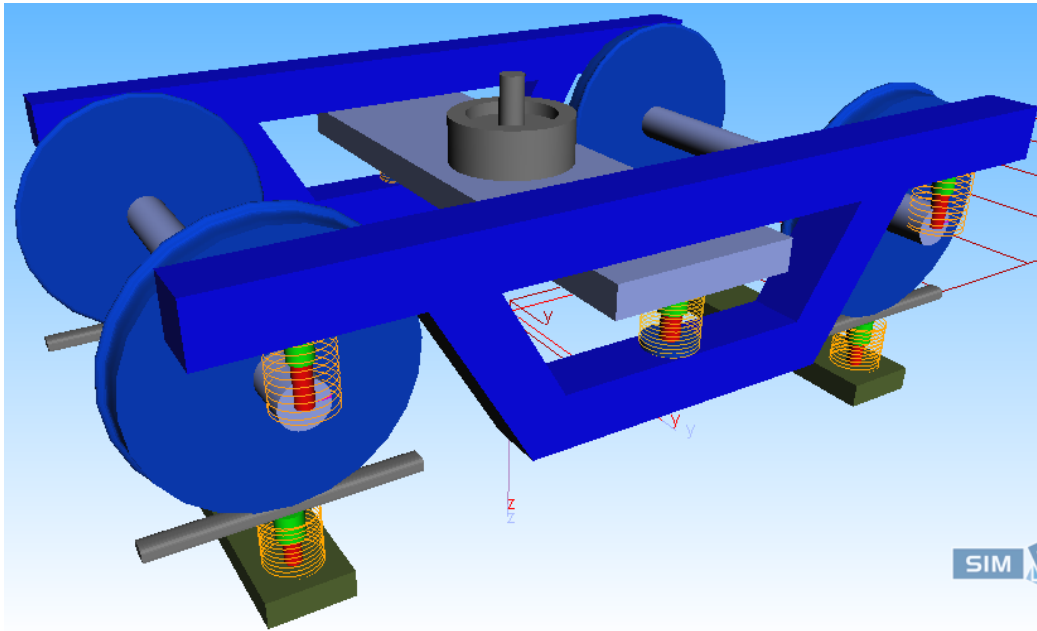
Figure 3.13 (a) Vertical primary damping coefficient (b) secondary damping coefficient

3.4.4 Inserting track stiffness and damping (Flexibility of the track)

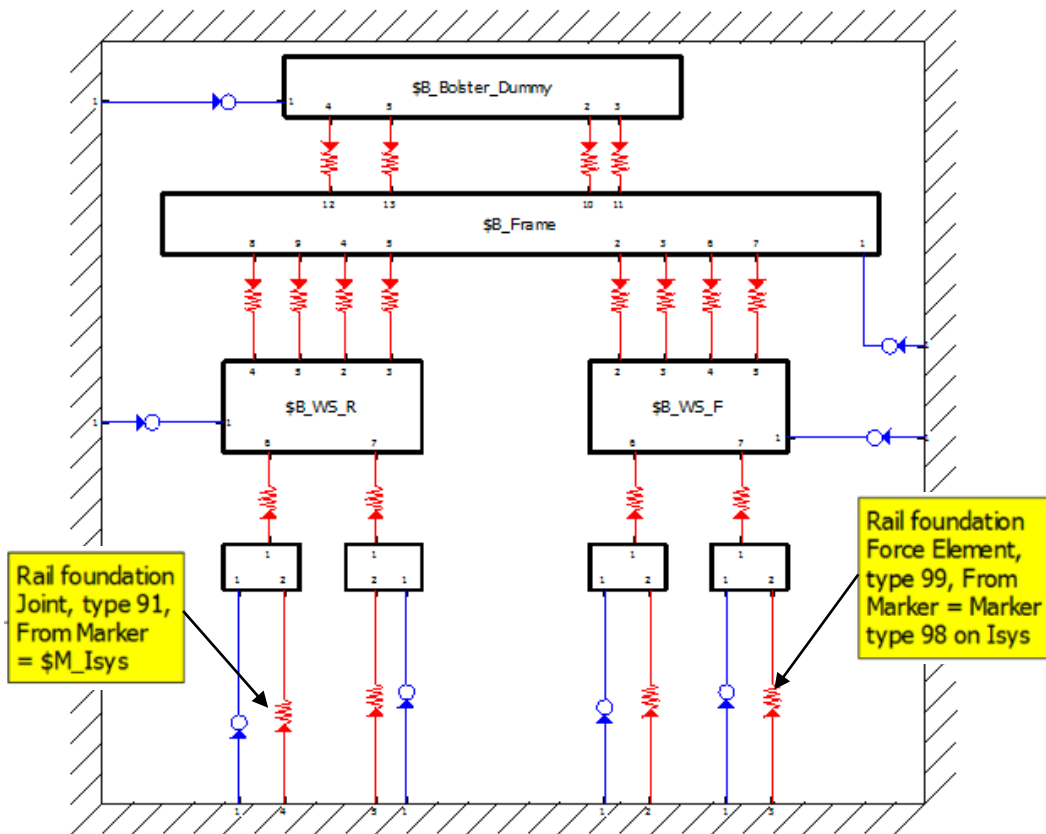
Insert two markers of type 98: Follow Track Joint one for each wheel and insert two primitives of type 1: Cuboid using the reference markers created to represent the ground. Create markers of type 98: Follow Track Joint on the attachment points of forces on the primitive and the rail. Insert force type 98: Rail Track Ballast between the markers using the data given in Table 3.6 and create the primitives following the same way as the primary suspension. Save the model. The bogie with the track stiffness model is shown in Figure 3.14. SIMPACK GUI used to insert the values is shown in Appendix B.

Table 3.6 Stiffness and damping values for the track [22]

Parameter	Value
Constant spring coefficient about y	20 MN/m
Constant spring coefficient about z	75 MN/m
Constant spring coefficient about x	15 MN/m
Constant damper coefficient in y	45000Ns/m
Constant damper coefficient in z	60000Ns/m
Constant damper coefficient in x	30000 Ns/m
Spring coefficient amplitude in y	3210 N/m
Spring coefficient amplitude in z	6340 N/m
Spring coefficient amplitude in x	2540 N/m
Damper coefficient amplitude in y	3210 Ns/m
Damper coefficient amplitude in z	4450 Ns/m
Damper coefficient amplitude in x	2310 Ns/m
Sleeper spacing	0.7m



(a)



(b)

Figure 3.14 Model of bogie with flexible track (a) 3D and (b) 2D model

3.4.5 Creating car body

Open new 'General' template and adjust axis directions. Change the body type 21: Wheel Rail Cab and insert the appropriate dimensions. Insert center plate attachment cylinders and on the center of these cylinders insert a marker of type 1: Identity Matrix. Change the joint type 9: Rheonomic Rail Track Joint.

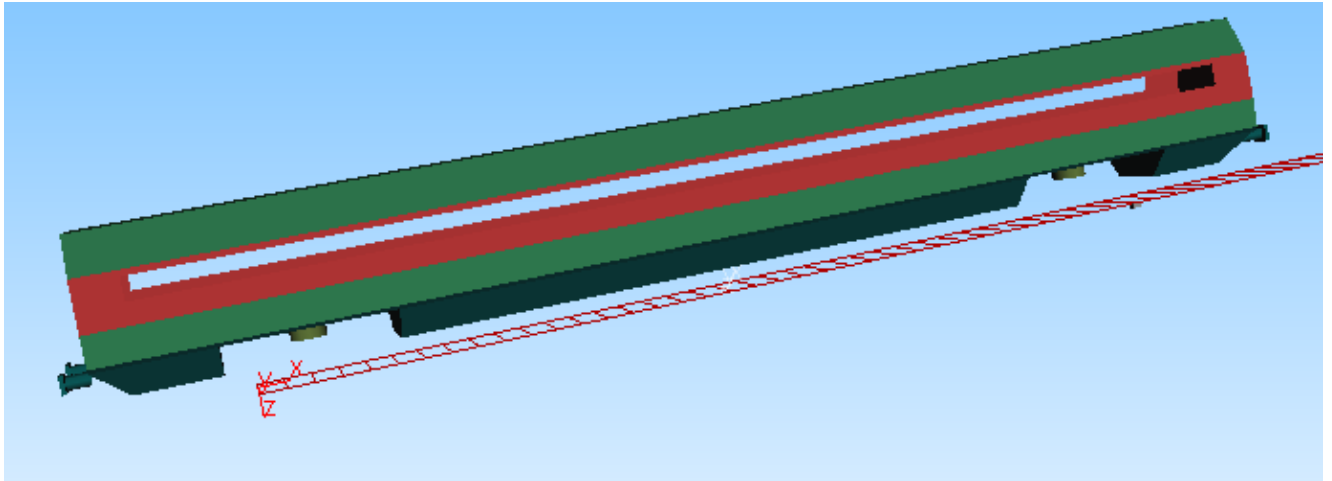
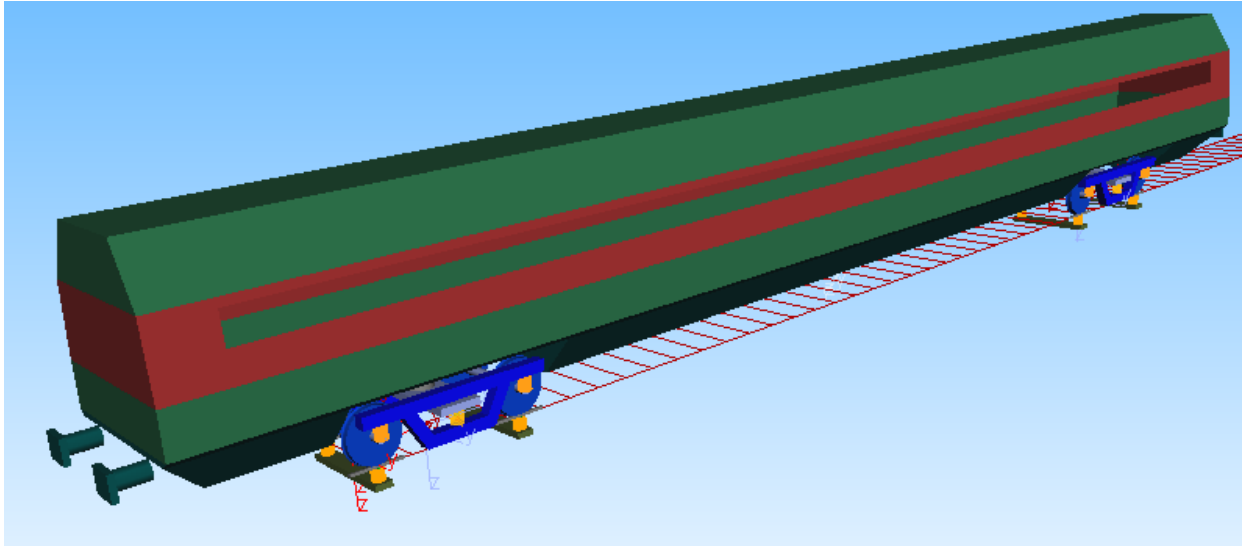


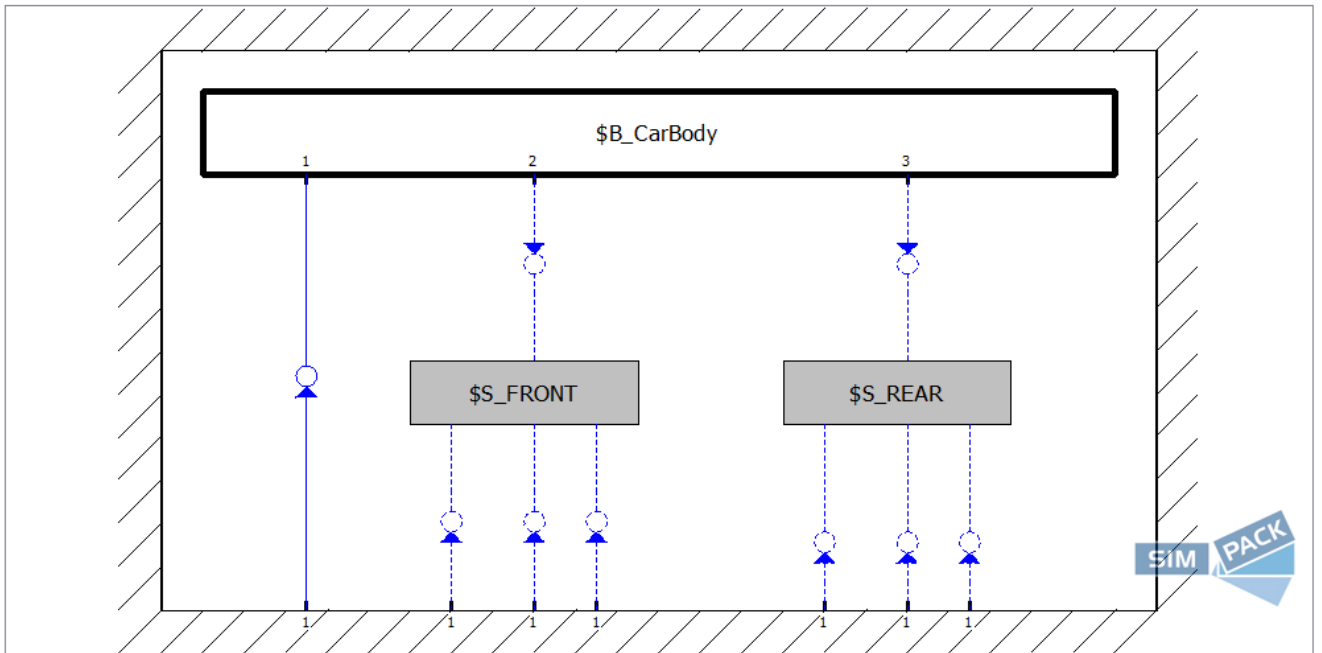
Figure 3.15 Car body model

3.4.6 Finalizing the model

Insert the bogie model as a substructure to the car body model two times one for the front and the other for the rear. Use the markers created on the bolster and car body center plate to attach the bogies on the respective positions. Create Input Function for the speed profile. Create excitation of type 20: Speed Profile Along Track and insert the speed profile function. Insert sensors where data recording is needed. Perform preload calculation in order to bring the model to initial equilibrium. The final model is shown in Figure 3.16.



(a)



(b)

Figure 3.16 Final vehicle SIMPACK model (a) 3D and (b) 2D model.

CHAPTER FOUR

MODEL VALIDATION, RESULT AND DISCUSSION

In this section, the developed SIMPACK model will be validated and the output results will be presented and discussed. Since, the purpose of the study is to analyze the effect of track stiffness variation on the dynamic behavior of the vehicle, two models are developed: one model with constant track stiffness and damping and the other variable. The stiffness and damping variability applied in the model by giving sinusoidal varying stiffness and damping inputs in addition to the constants. The amplitudes are given in Table 3.5 and the mathematical force equations are given in section 3.2.2.

4.1 Model validation

An adequate representation of a real vehicle–track system through a MBS model requires a proven computer tool which able to represent the phenomena under investigation and an application of appropriate modeling technique. SIMPACK software, which is used in this study, is widely accepted and proven for dynamic analysis of vehicles. Even if SIMPACK is a proven tool, the modeling has to be carried out carefully to avoid mistakes and errors. So, the developed model needs to be checked and tested before being used for the assessment of vehicle performance. In this research, plausibility check will be done.

Plausibility check

The flexibility of the track was included in the model by equivalent constant stiffness and damping and also it was made to vary sinusoidal with the wave length equal to the sleeper spacing and amplitude equal to the stiffness and damping variation. The input data for modeling the flexibility of the track is given in Table 3.6. After performing the analysis the output stiffness and damping should look like the input data. Figure 4.1 shows the stiffness output and Figure 4.2 shows the damping output in the lateral and vertical direction. These outputs are compatible with the input. As given to the software the variation of stiffness and damping begins after 50 meter, this is clearly seen in the output also. In addition, Figure 4.1 to 4.4 shows the outputs are compatible with the inputs for the variable and constant stiffness and damping cases.

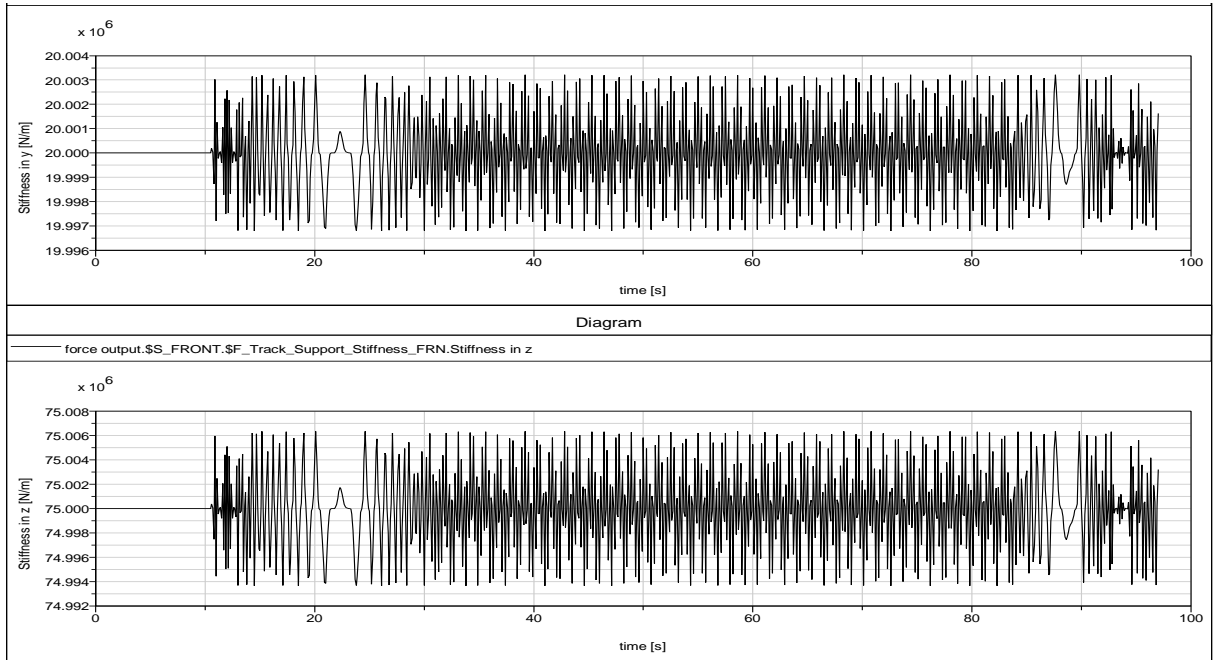


Figure 4.1 Track stiffness output in the lateral and vertical direction for the variable case

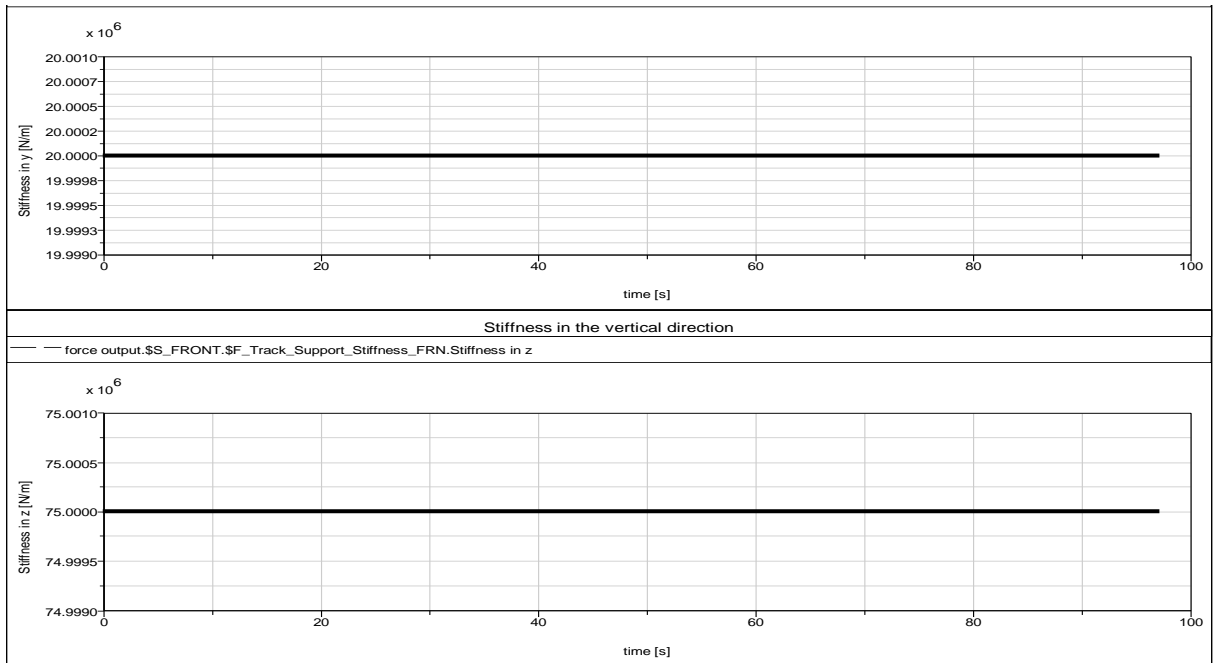


Figure 4.2 Track stiffness output in the lateral and vertical direction for the constant case

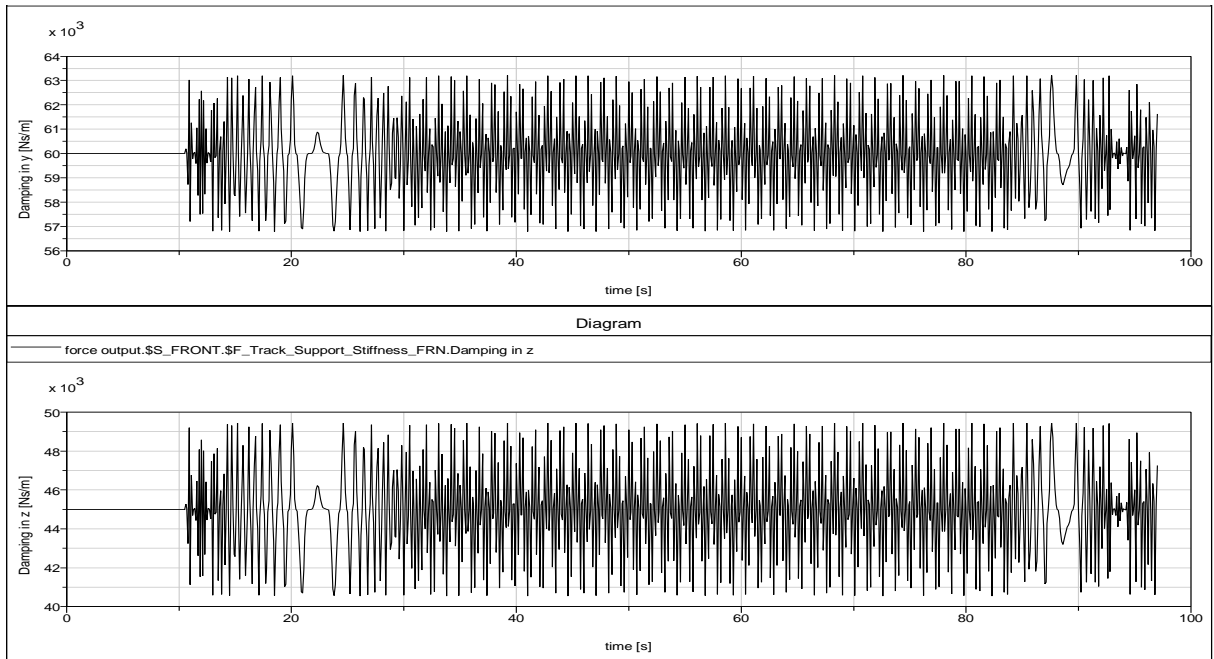


Figure 4.3 Track damping output in the lateral and vertical direction for the variable case

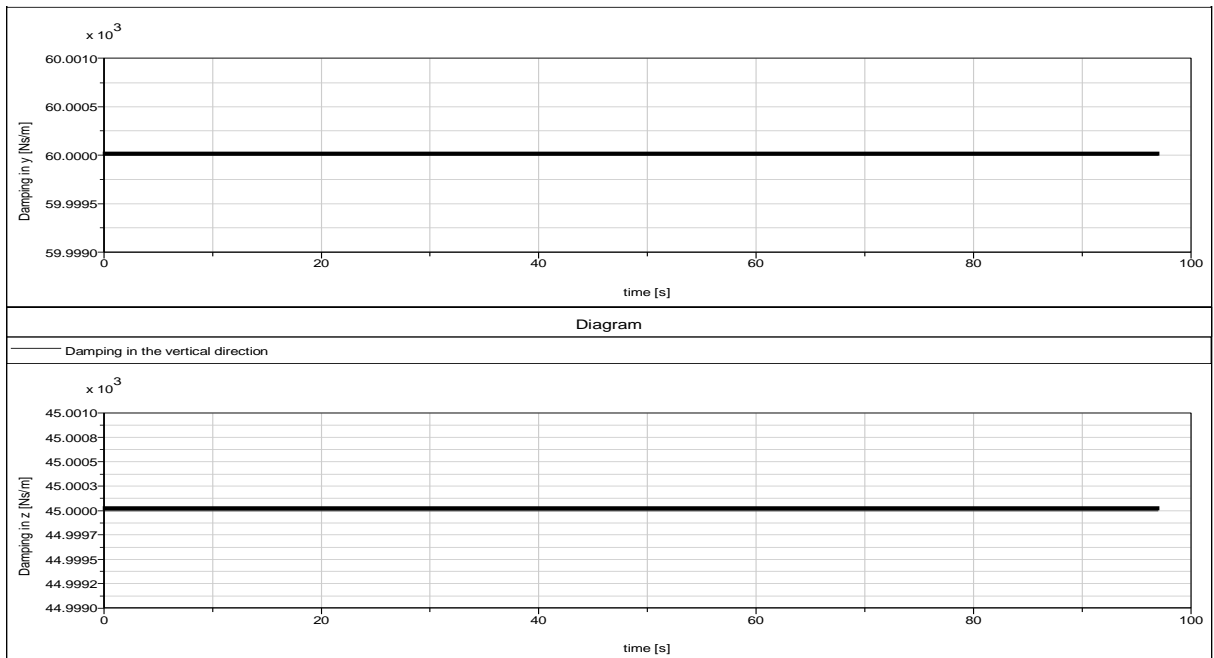


Figure 4.4 Track damping output in the lateral and vertical direction for the constant case

4.2 Result and discussion

The dynamic simulation output values depend on motion of the vehicle i.e. its velocity. So, to understand the effect of track stiffness variation on the vehicle the same model was analyzed based on three different velocity profiles i.e. the motion of the vehicle during service braking, emergency braking and taking greater acceleration and deceleration values. The standard for the velocity profiles are taken from technical specification. [26]

Table 4.1 Acceleration and deceleration values for emergency and service braking cases. [26]

Description	Speed range	Acceleration	Deceleration
Acceleration	0 – 40 Km/hr (0-11.11 m/s)	$\geq 1 \text{ m/s}^2$	-
Acceleration	40 – 70 Km/hr (11.11-19.44 m/s)	$\geq 0.5 \text{ m/s}^2$	-
Coasting	70 – 50 Km/hr (19.44-13.89 m/s)	-	Assumed
Deceleration for service braking	50 – 0 Km/hr (13.89-0 m/s)	-	$\geq 1.1 \text{ m/s}^2$
Deceleration for emergency braking	70 – 0 Km/hr (19.44-0 m/s)	-	$\geq 2 \text{ m/s}^2$

4.2.1 Dynamic effect of track stiffness variation during service braking (SB)

During service braking the driver stops power input to the vehicle for some time leading to decelerate because of different resistances without the application of the brake. Then braking will be applied. The velocity profile followed by the vehicle during service braking is given in figure 4.5.

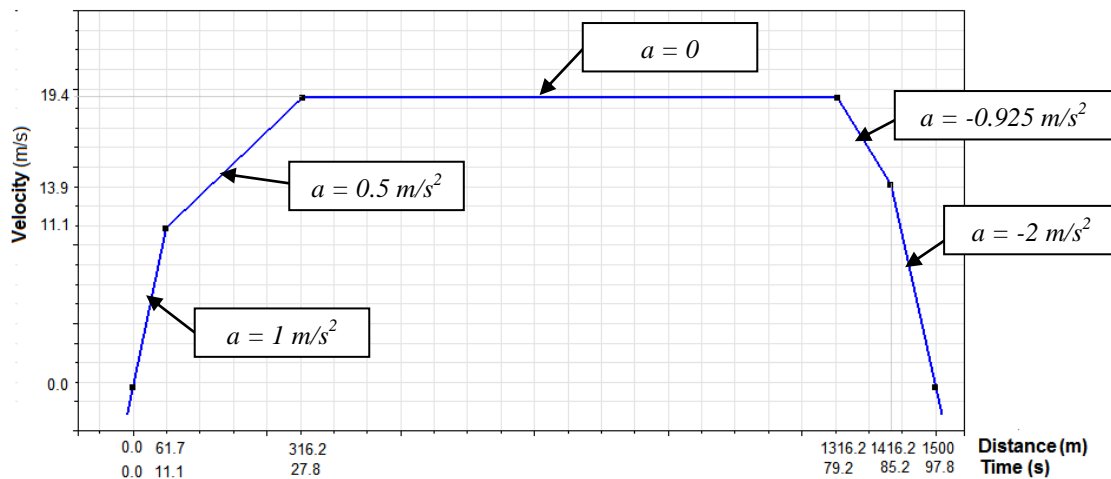


Figure 4.5 Velocity profile curve followed by the vehicle during service braking (SB)

4.2.1.1 Effect of track stiffness variation on car body displacement

Figure 4.6 and 4.7 show the time histories of the displacement of the car body in the vertical direction for the constant and variable stiffness cases respectively. As it can be seen for both cases, when the vehicle is at rest the car body had moved downwards from its original position this is because the car body load had compressed the suspension system during the preload calculation. During the starting of the motion, the car body moved up and down for short period and settled. The largest displacement for both cases occurred when the vehicle is coasting i.e. cutting power supply to the vehicle. However, comparing the values of the two cases, the variable track stiffness case has showed more vertical displacement. The two cases had showed similarity in the vertical displacement along the motion of the vehicle. During brake application, the variable track stiffness case had showed greater displacement increase.

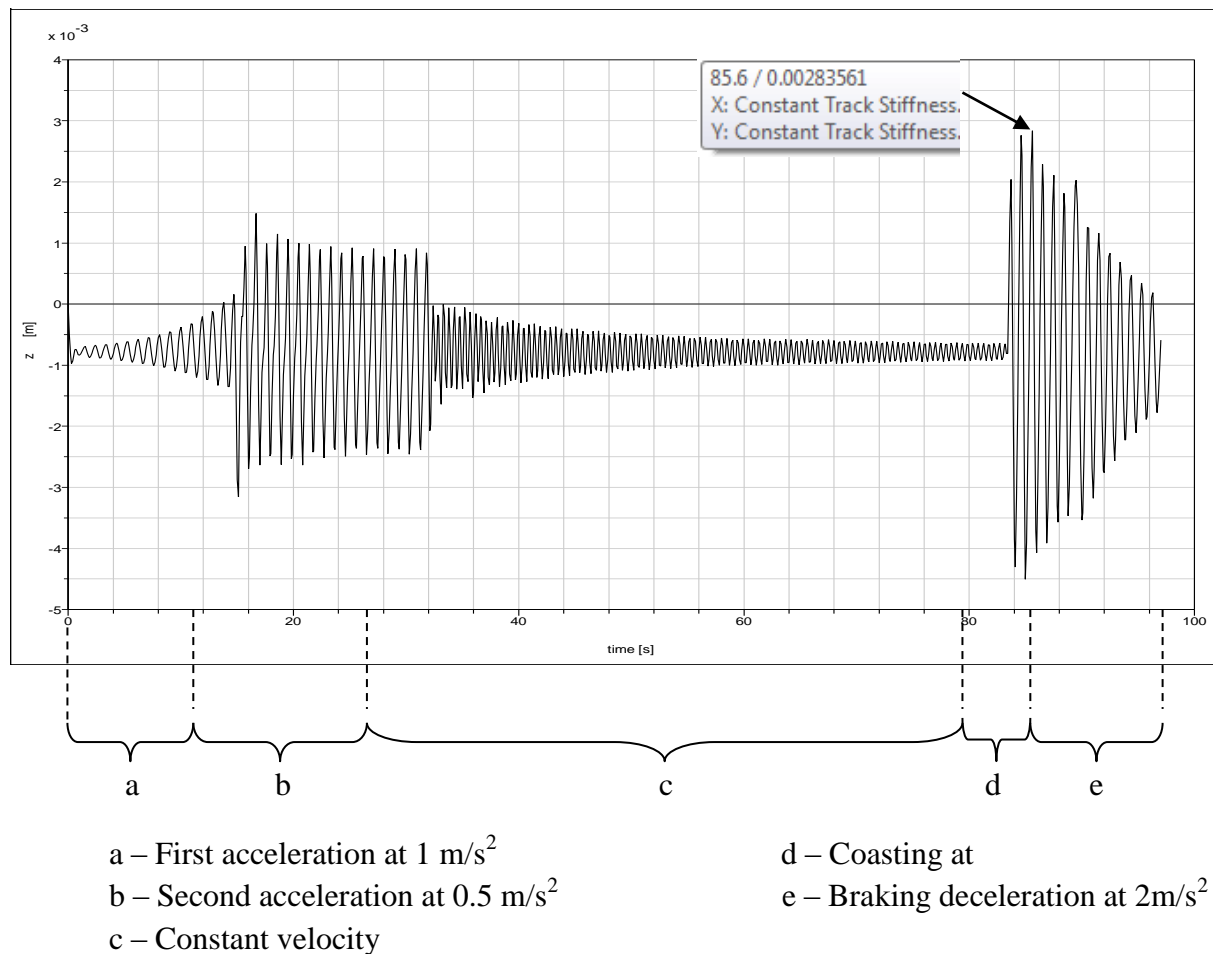


Figure 4.6 Vertical displacement of the car body with constant track stiffness during SB

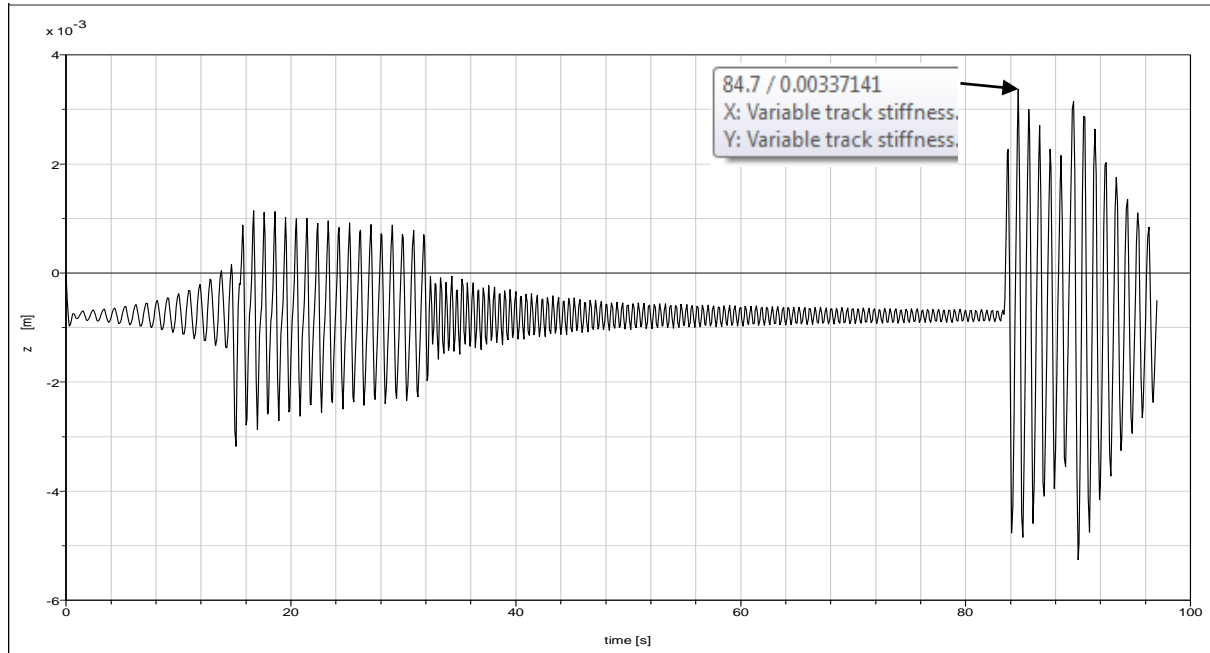


Figure 4.7 Vertical displacement of the car body with variable track stiffness during SB

Figure 4.8 and 4.9 show the lateral displacement of the car body for the constant track and variable track stiffness respectively. Before starting the motion, i.e. after the preload calculation, the car body was at zero position laterally. For both cases, the change of lateral displacement of the car body is higher after the change of the longitudinal acceleration of the vehicle. In the case of constant track stiffness, the lateral displacement dies very soon and its value is almost zero during coasting and deceleration. When the vehicle moves on the variable stiffness track, the lateral displacement of the car body is higher after the first and second change of acceleration. In addition to this, the car body showed some lateral displacement during decelerating. For the two cases, the peak values occurred at different locations i.e. at 37.2 second for constant track stiffness and 24.8 second for the variable track stiffness.

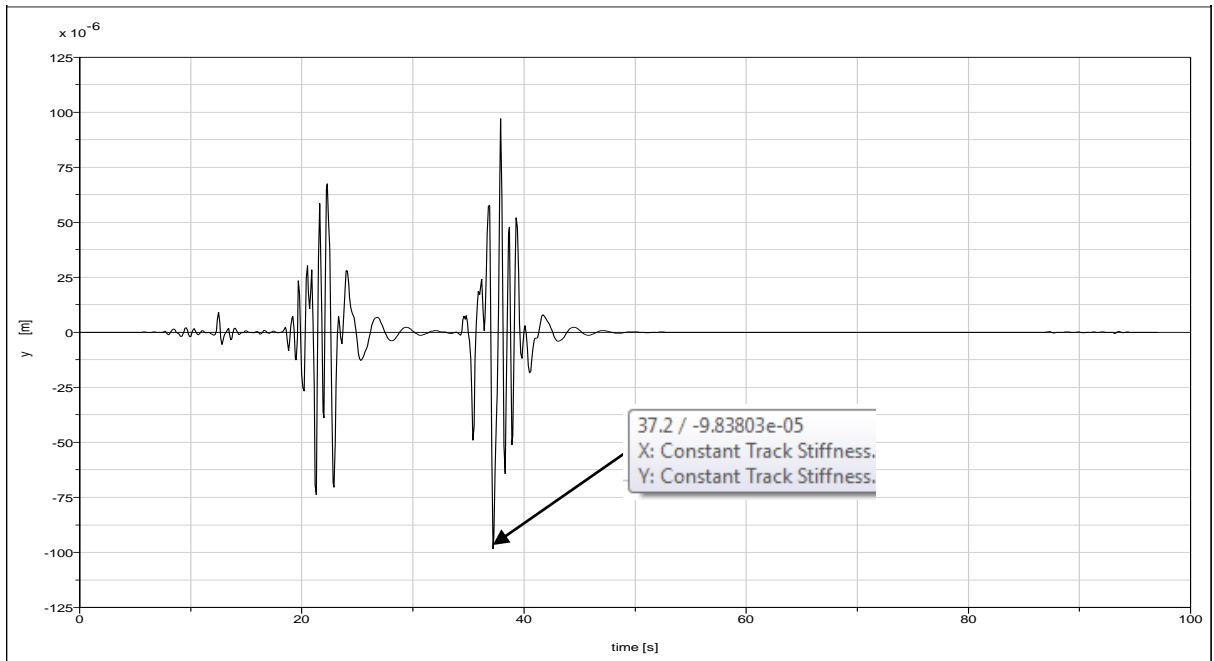


Figure 4.8 Lateral displacement of the car body with constant track stiffness during SB

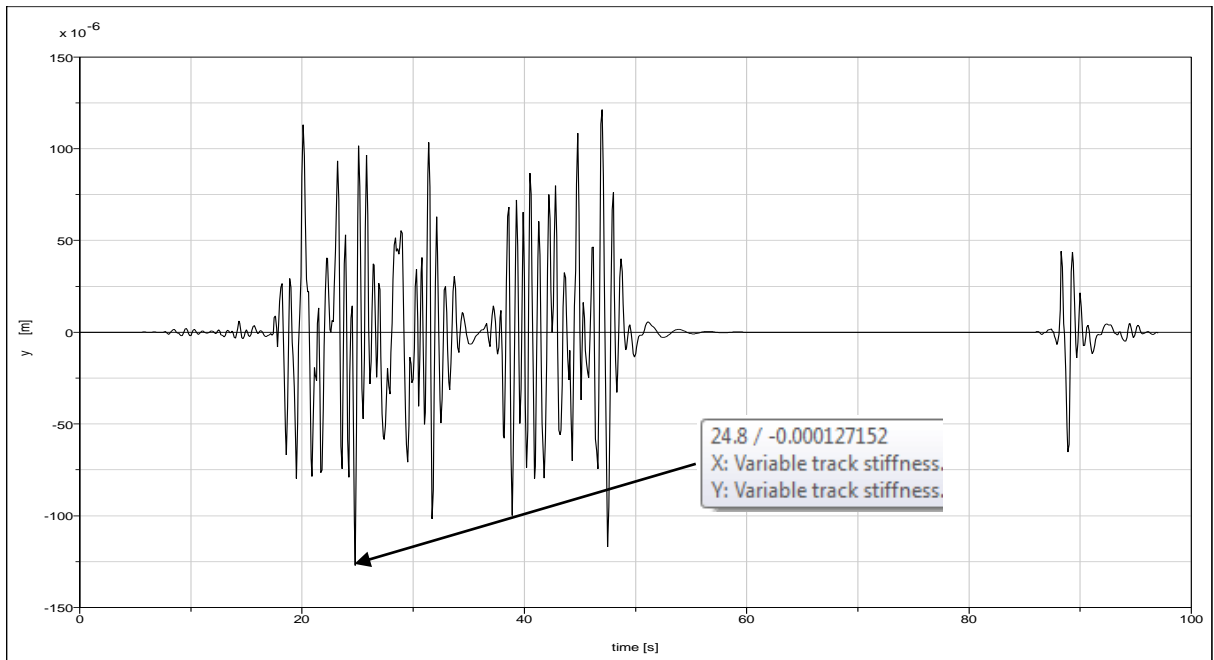


Figure 4.9 Lateral displacement of the car body with variable track stiffness during SB

4.2.1.2 Effect of track stiffness variation on car body acceleration

Before starting the motion the vertical acceleration of the car body is zero. This shows the preload calculation is made correctly with no residual acceleration. In both cases, an increment in the vertical acceleration of the car body was showed when the vehicle changes its longitudinal acceleration. After the increment, the vertical acceleration of the car body decreased continuously along the motion of the vehicle for both cases.

Figures 4.10 and 4.11 show car body vertical acceleration for the constant and variable track stiffness cases respectively. As it is shown in the figures, both cases showed largest vertical acceleration at 85 second, however the value for variable track stiffness is greater.

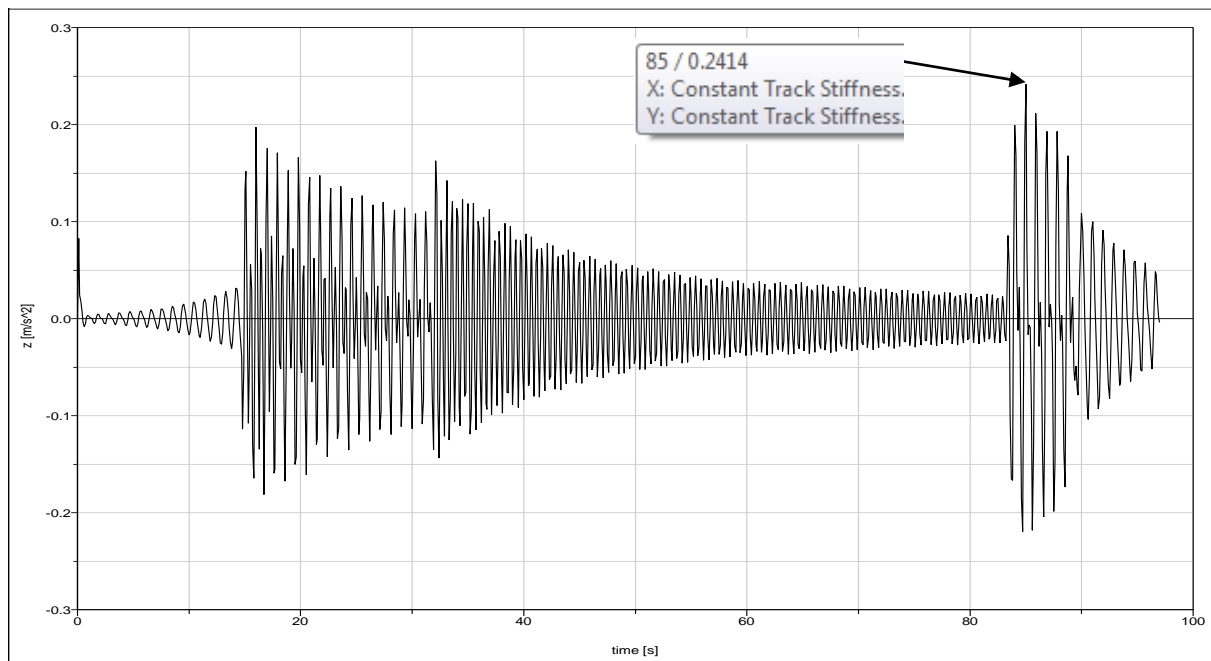


Figure 4.10 Vertical acceleration of the car body with constant track stiffness during SB

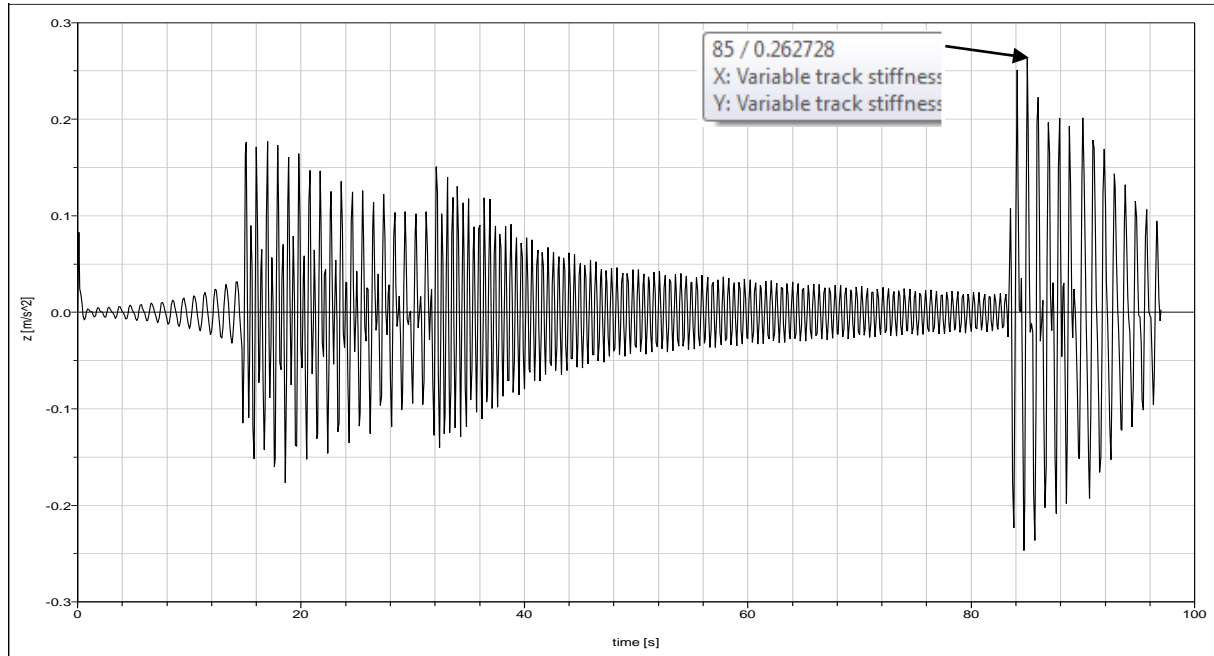


Figure 4.11 Vertical acceleration of the car body with variable track stiffness during SB

In the same way, before starting the motion the lateral acceleration of the car body was zero. Figure 4.12 and 4.13 show the lateral acceleration of the car body. For both cases, the car body showed erratic lateral acceleration after changing the first and the second longitudinal acceleration of the vehicle. However, the lateral acceleration for the constant track stiffness case dies soon compared to the variable track stiffness case. In addition, the car body showed small lateral acceleration during deceleration in the case of variable track stiffness. The peak values for the two cases occurred at different locations i.e. at 21.9 second for constant track stiffness and 44.1 second for the variable track stiffness. Comparing the two outputs the variable track stiffness case has greater peak lateral acceleration value.

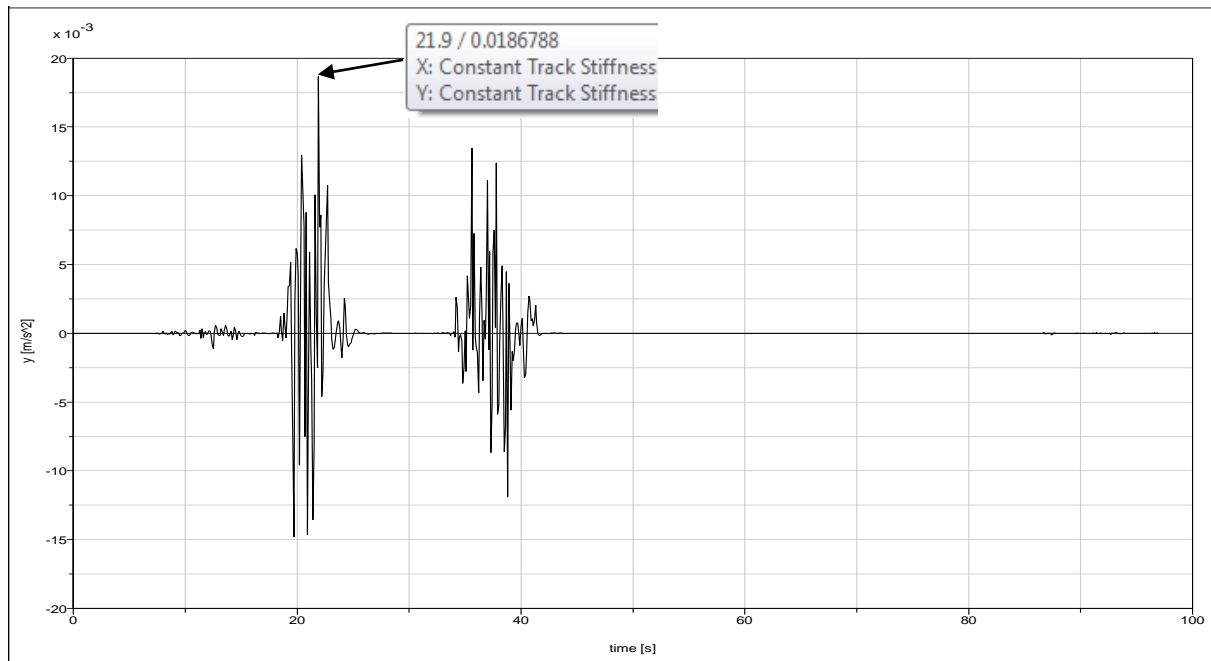


Figure 4.12 Lateral acceleration of the car body with constant stiffness during SB

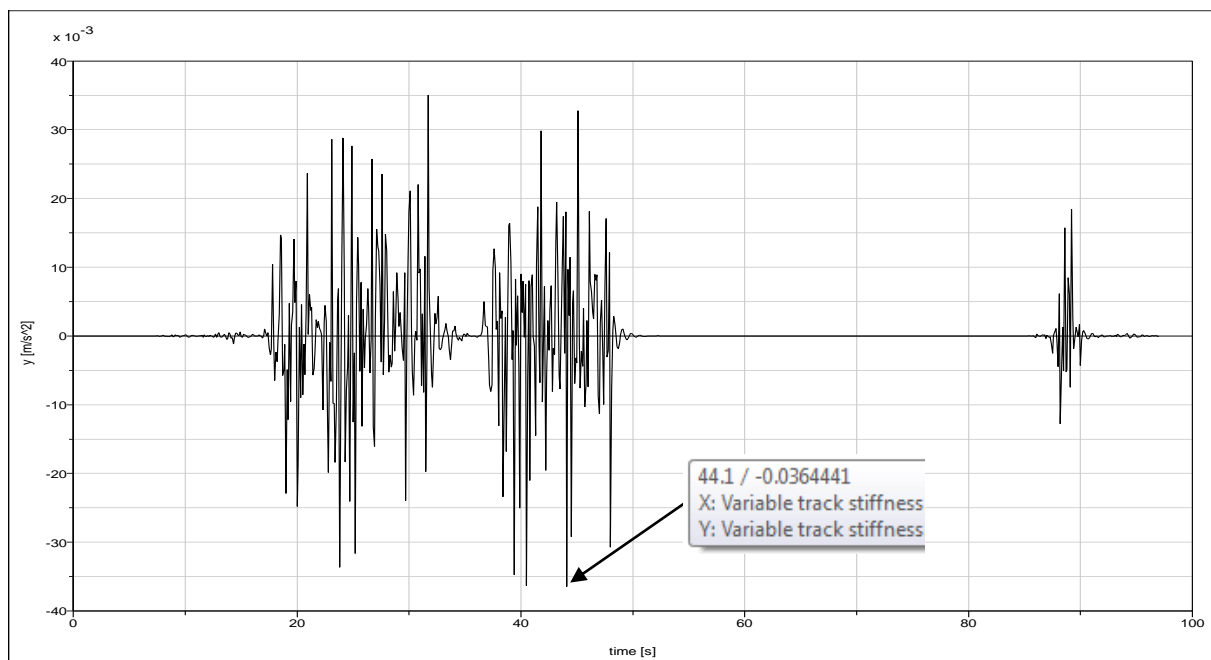


Figure 4.13 Lateral acceleration of the car body with variable stiffness during SB

4.2.1.3 Effect of track stiffness variation on bogie frame displacement

The bogie frame vertical displacement is shown in Figure 4.14 and 4.15. As it can be seen from the output, at static equilibrium the frame moved down for both cases. This is because of the preload calculation. Again for both cases, the frame moved down upon vibrating during longitudinal acceleration and deceleration. At the constant velocity section the frame gets back to its original position for both cases. An increment in the vertical displacement occurred during the change of longitudinal acceleration of the vehicle for both cases. After the increment however, the vertical displacement reduces continuously along the motion of the vehicle. The largest vertical frame displacement occurred around 90 second for both cases. As shown in the figures the variable stiffness case has greater peak vertical displacement value.

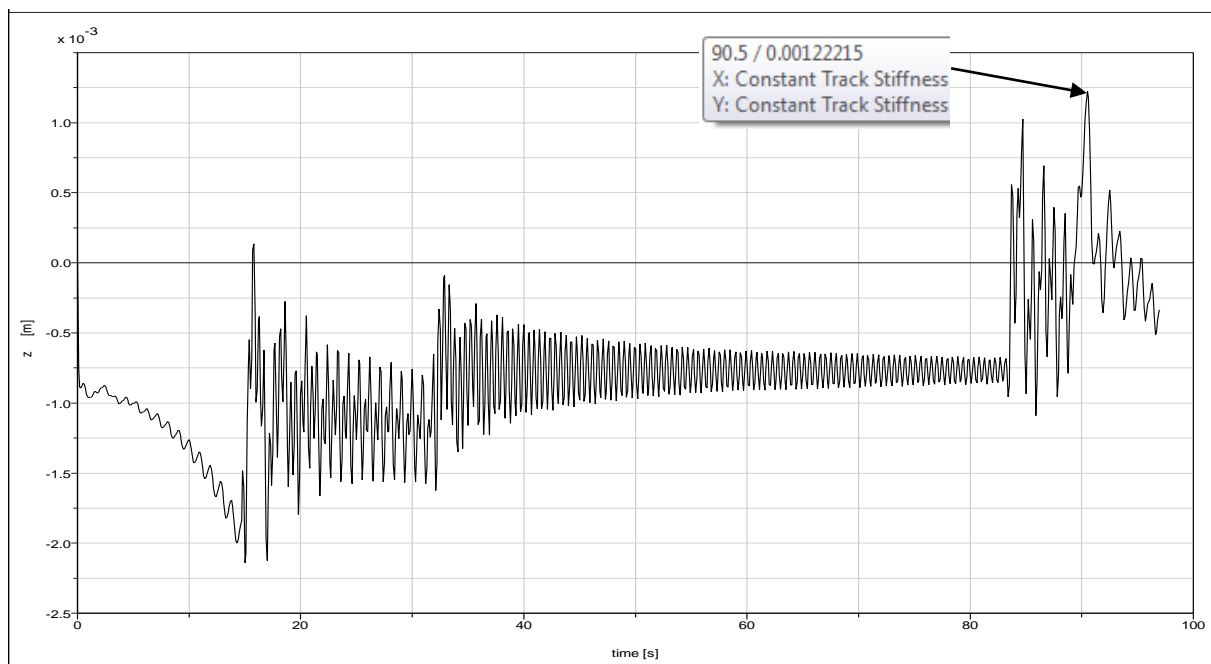


Figure 4.14 Vertical displacement of the frame with constant stiffness during SB

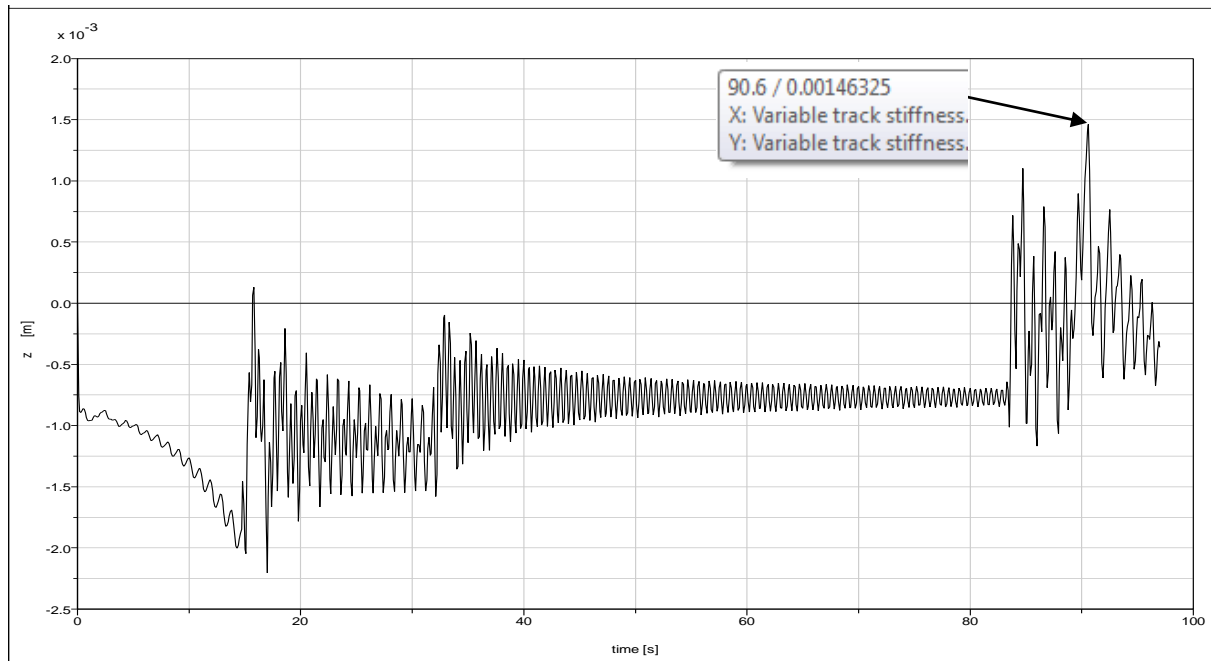


Figure 4.15 Vertical displacement of the frame with variable stiffness during SB

Figure 4.16 and 4.17 shows the lateral displacement of the frame. The frame was at zero position before starting the motion for both cases. An increment in the lateral displacement was shown during the vehicle changes its longitudinal acceleration. The motion of the frame in the lateral direction was very small for both cases. During longitudinal deceleration, the variable track stiffness case shows small lateral displacement. Comparing the peak values the constant track stiffness case has more value.

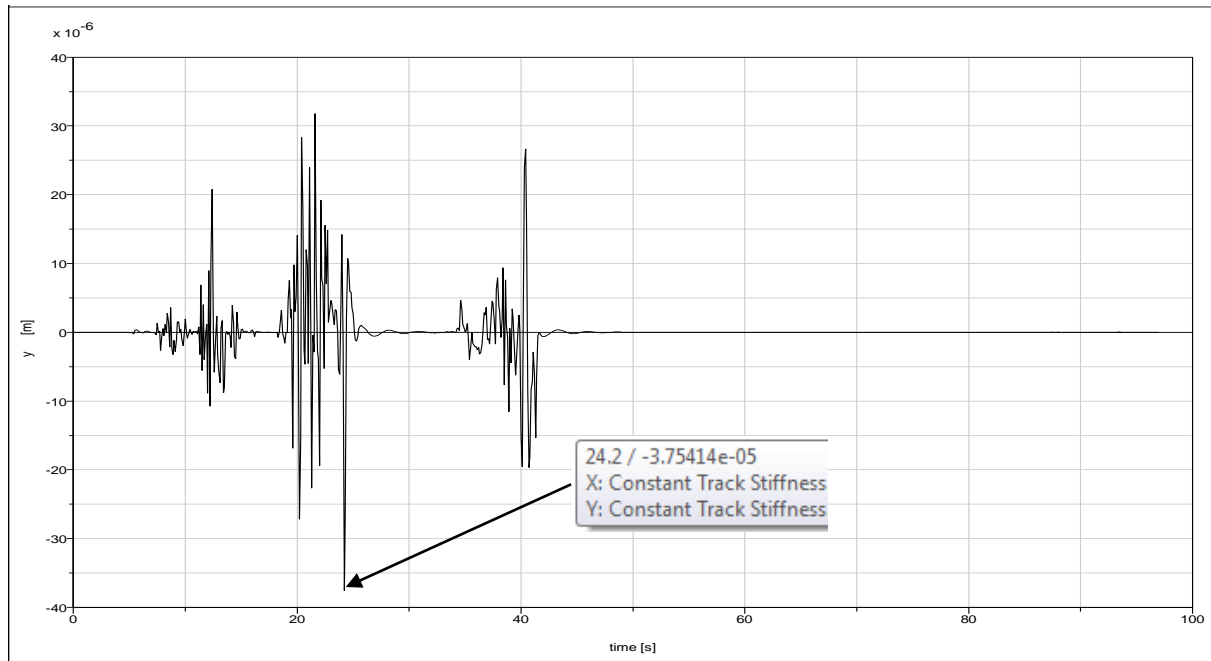


Figure 4.16 Lateral displacement of the frame with constant track stiffness during SB

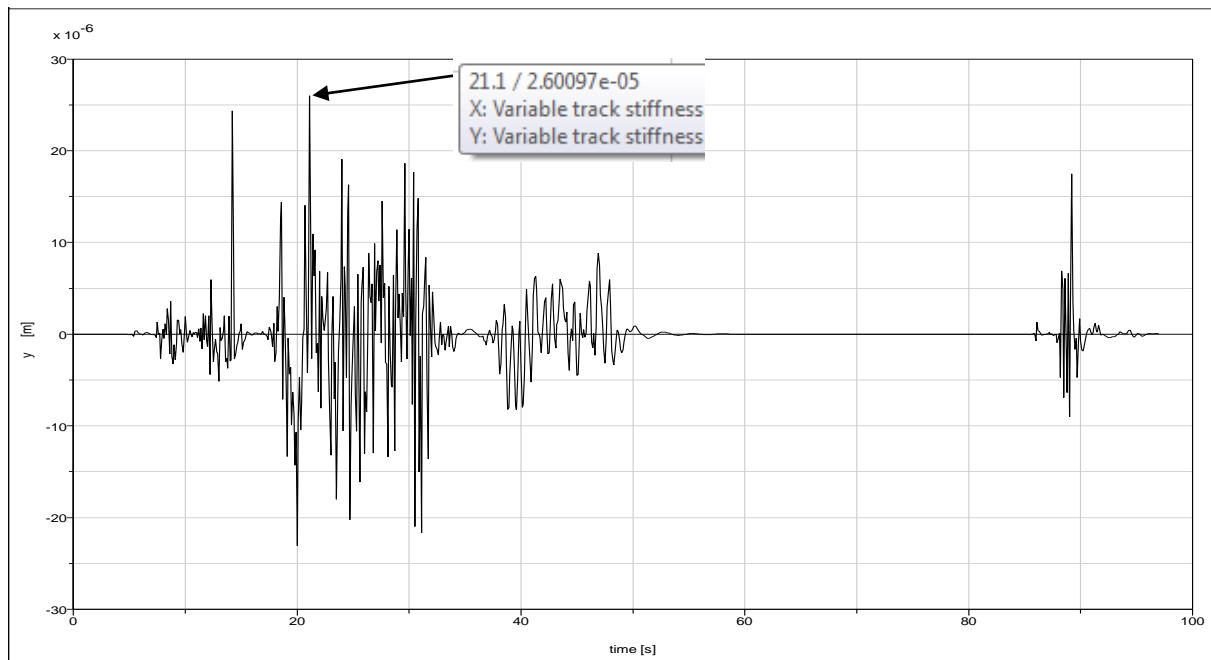


Figure 4.17 Lateral displacement of the frame with variable track stiffness during SB

4.2.1.4 Effect of track stiffness variation on bogie frame acceleration

As it can be seen from Figure 4.18 and 4.19, at the start of the motion the vertical acceleration of the frame for both cases is greater. This is because the vehicle starts motion. For both cases, higher vertical acceleration of the frame occurred at the second acceleration, beginning of constant velocity and when the vehicle is coasting. Comparing the two cases, the constant track stiffness case has greater vertical acceleration during the starting of the constant velocity motion of the vehicle and the variable track stiffness case has greater value during the second acceleration and coasting motion of the vehicle. However, if we take the peak of the two cases, the constant stiffness case has greater value as shown in Figure 4.18.

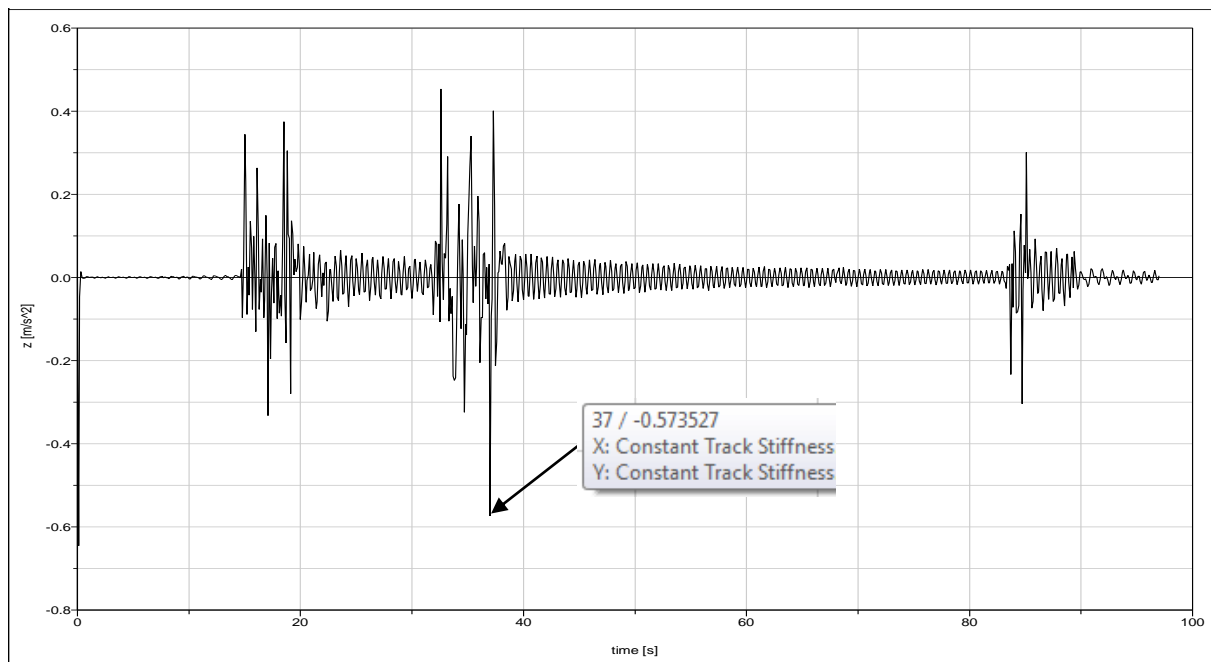


Figure 4.18 Vertical acceleration of the frame with constant stiffness during SB

Figure 4.20 and 4.21 show the lateral acceleration of the frame. The constant track stiffness case shows the lateral acceleration of the frame is higher during the second acceleration and on the starting of the constant velocity motion of the vehicle and it remains zero for the rest. On the other hand, the variable track stiffness case shows an erratic lateral acceleration on the second acceleration and on the starting of the constant velocity motion of the vehicle and it shows some lateral acceleration during coasting. Comparing the peak values of the two cases the constant track stiffness case has greater value.

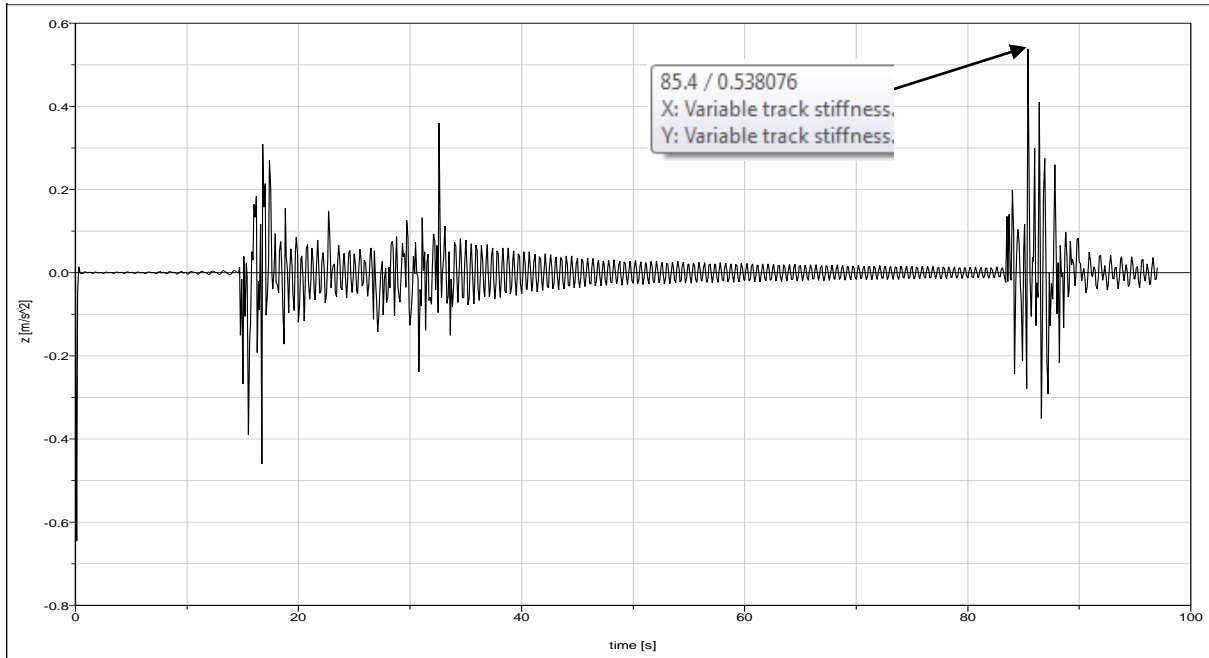


Figure 4.19 Vertical acceleration of the frame with variable stiffness during SB

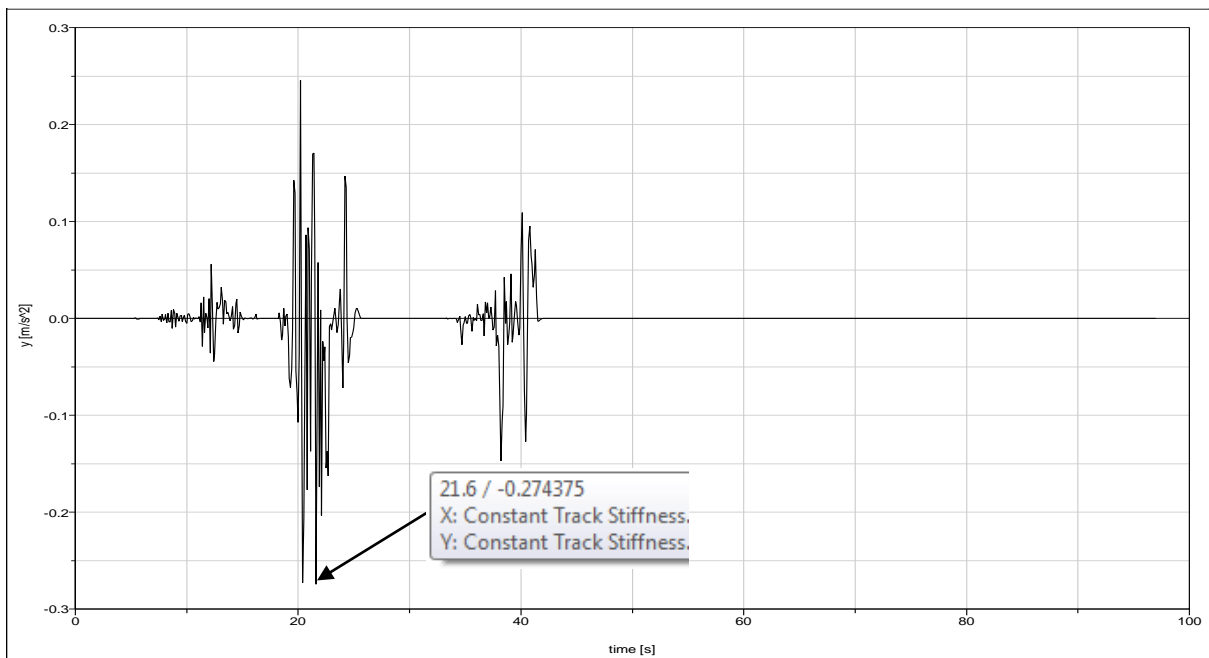


Figure 4.20 Lateral acceleration of the frame for the constant stiffness during SB

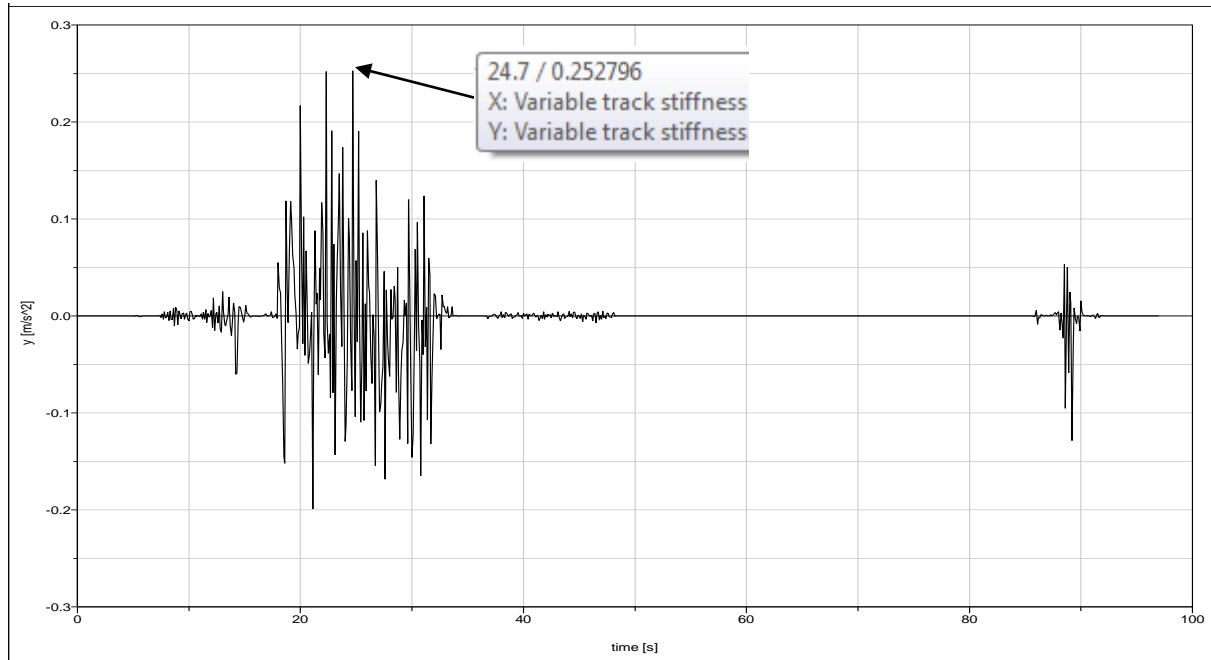


Figure 4.21 Lateral acceleration of the frame for the variable stiffness during SB

4.2.1.5 Summary of the outputs during service braking

As it can be seen from Table 4.2, the car body displacement and acceleration increased during the variable track stiffness. For the bogie frame however, it is not uniform i.e. the vertical displacement showed increment and the other values decreased during the variable track stiffness case.

Table 4.2 Summary of the output values for the constant and variable track stiffness during service braking motion.

Component	Measured quantity	Constant track stiffness	Variable track stiffness	% Increase or Decrease
Car body	Vertical displacement	0.00283561	0.00337141	18.9
	Lateral displacement	0.00009838	0.00012715	29.2
	Vertical acceleration	0.2414	0.262728	8.8
	Lateral acceleration	0.0186788	0.0364441	95.1
Bogie frame	Vertical displacement	0.0012221	0.00146325	19.7
	Lateral displacement	0.0000375	0.000026	-30.7
	Vertical acceleration	0.573527	0.538076	-6.2
	Lateral acceleration	0.274375	0.252796	-7.9

4.2.2 Dynamic effect of track stiffness variation during emergency braking (EB)

During emergency braking the driver applies the brake to stop the vehicle based on the required stopping distance. In this case, the same velocity profile was used by removing coasting of the vehicle. The velocity profile followed by the vehicle during emergency braking is given in figure 4.22.

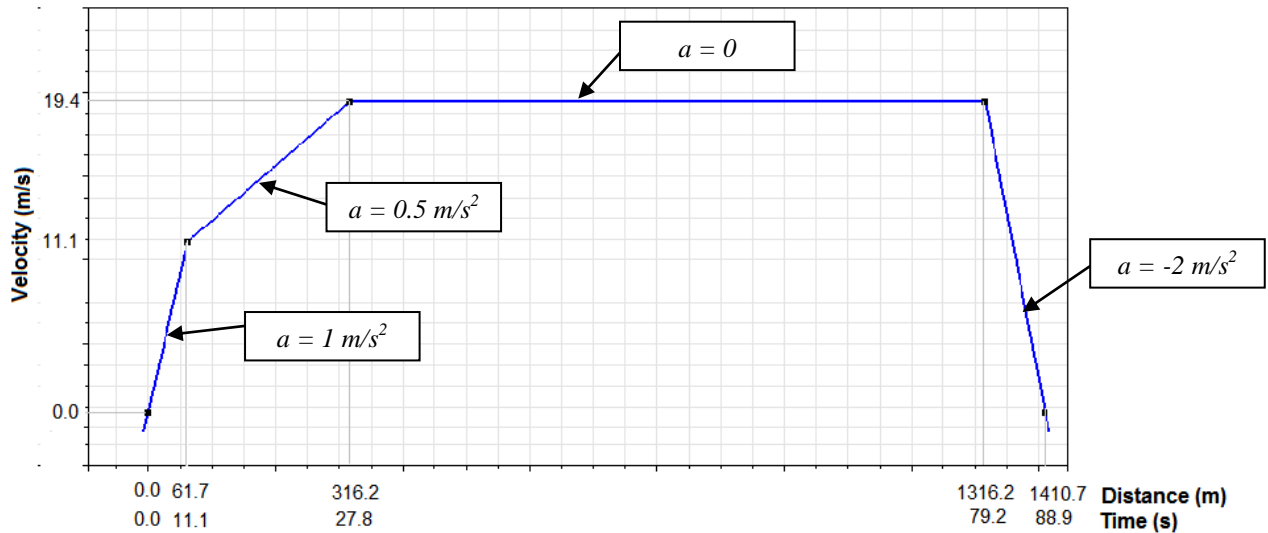


Figure 4.22 Velocity profile followed during emergency braking (EB)

4.2.2.1 Effect of track stiffness variation on car body displacement

Figure 4.23 and 4.24 show the time histories of the displacement of the car body in the vertical direction for the constant and variable stiffness cases respectively. As it can be seen for both cases, when the vehicle is at rest the car body had moved downwards from its original position this is because the car body load had compressed the suspension system during the preload calculation. The largest vertical displacement for both cases occurred around 5 seconds later after the application of the brake. However, comparing the values of the two cases, the variable track stiffness case has showed more vertical displacement. The two cases had showed similarity in the vertical displacement along the motion of the vehicle. Comparing the respective outputs of emergency braking with service braking, the emergency braking outputs are higher.

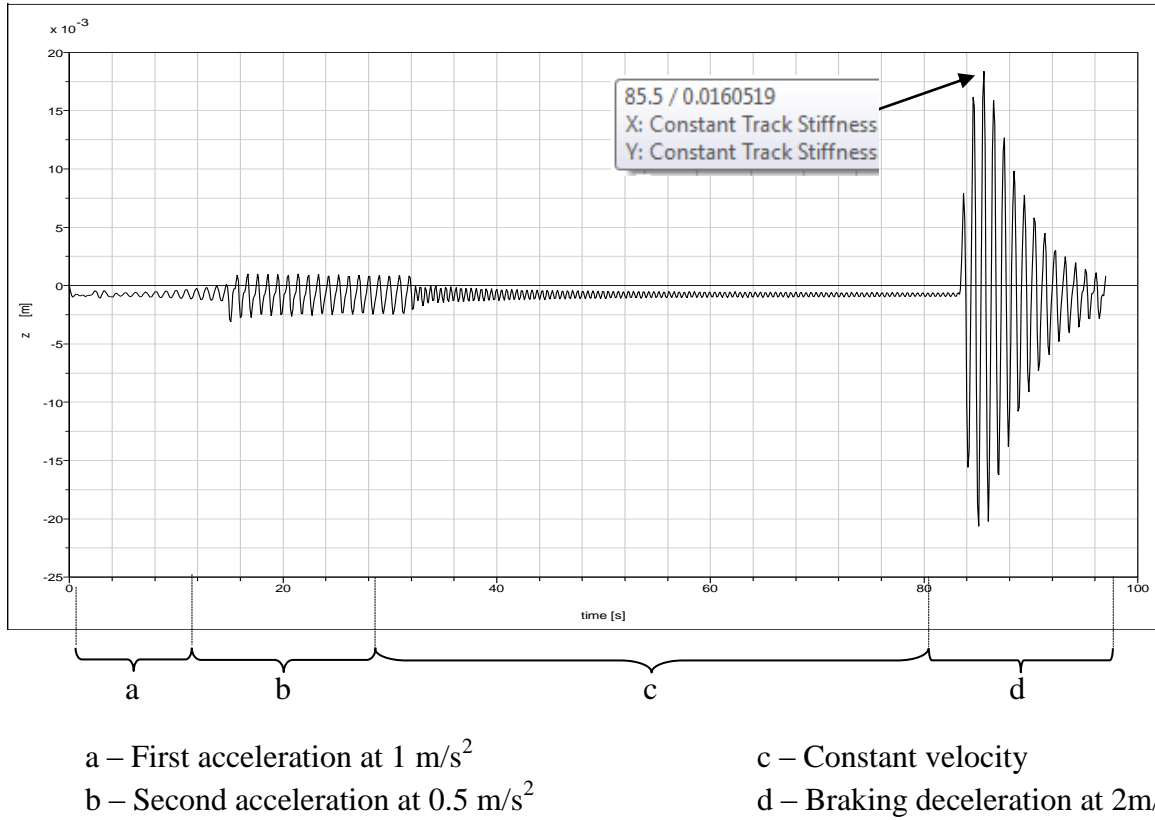


Figure 4.23 Vertical displacement of the car body with constant track stiffness during EB

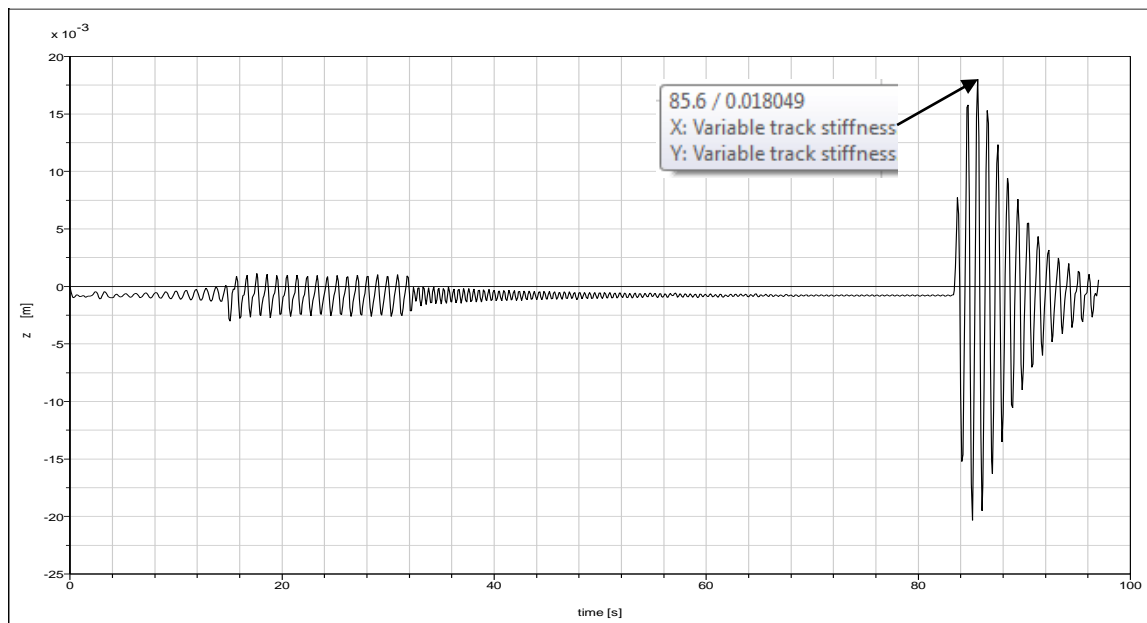


Figure 4.24 Vertical displacement of the car body with variable track stiffness during EB

Figure 4.25 and 4.26 show the lateral displacement of the car body for the constant track and variable track stiffness respectively. In the case of constant track stiffness, the lateral displacement dies very soon. When the vehicle moves on the variable stiffness track, the lateral displacement of the car body is higher during the constant velocity motion. Comparing the peak values of the two cases the variable track stiffness case showed greater lateral displacement.

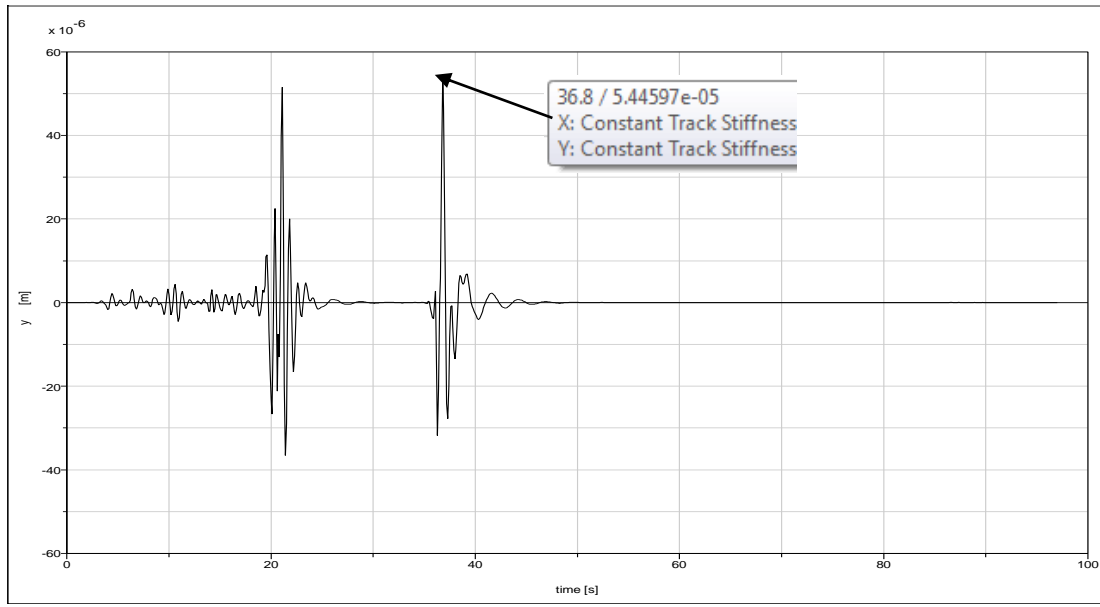


Figure 4.25 Lateral displacement of the car body with constant track stiffness during EB

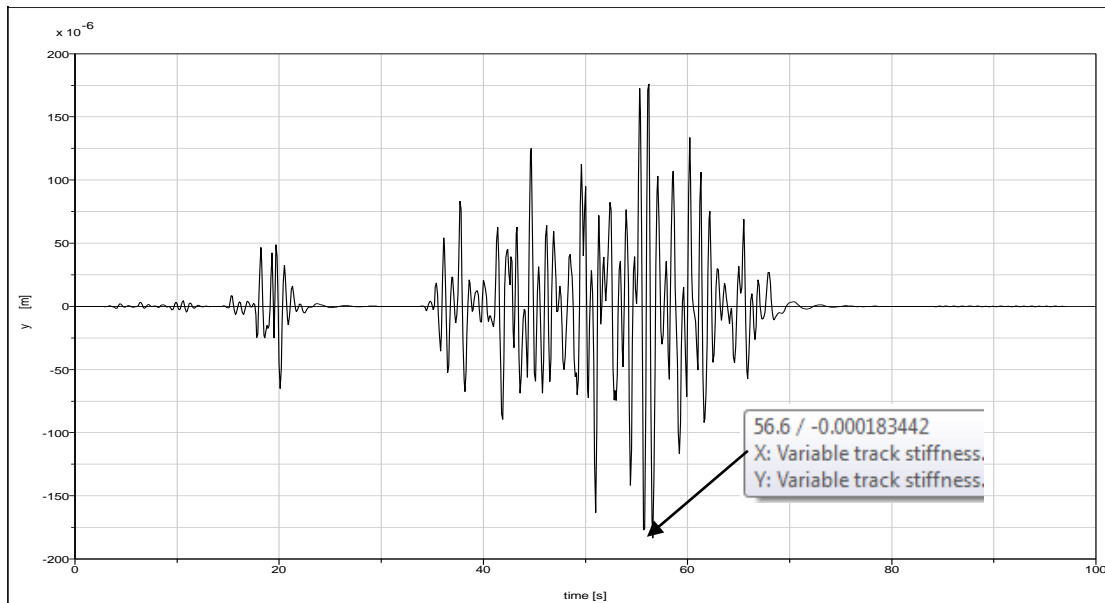


Figure 4.26 Lateral displacement of the car body with variable track stiffness during EB

4.2.2.2 Effect of track stiffness variation on car body acceleration

Before starting the motion the vertical acceleration of the car body is zero. This shows the preload calculation is made correctly with no residual acceleration. Figure 4.27 and 4.28 show car body vertical acceleration for the constant and variable case respectively. As it is shown in the figures, both cases showed largest vertical acceleration at 86 second, however the value for variable track stiffness is greater. Comparing the respective outputs, i.e. the variable and constant track stiffness, of the emergency and service braking cases, the emergency braking case has showed largest increase in vertical car body acceleration.

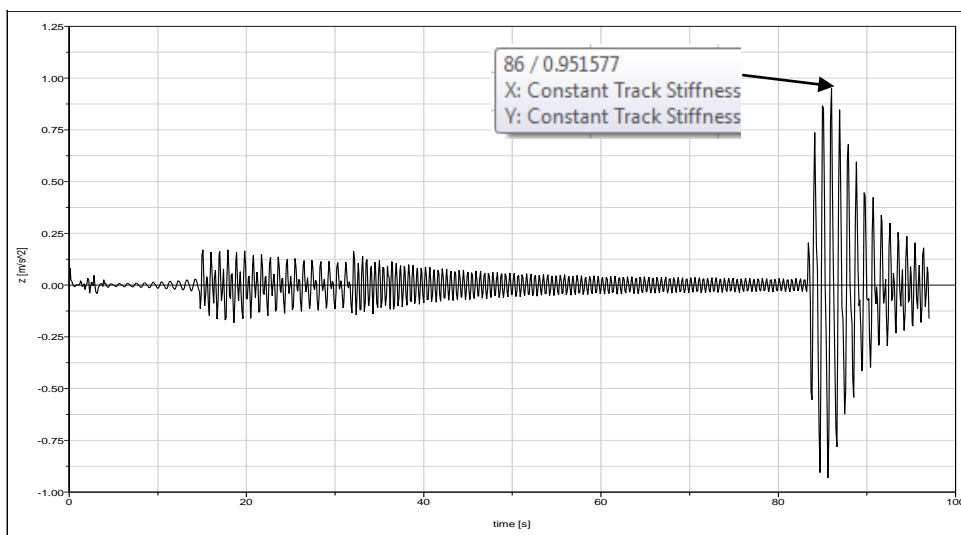


Figure 4.27 Vertical acceleration of the car body for constant track stiffness during EB

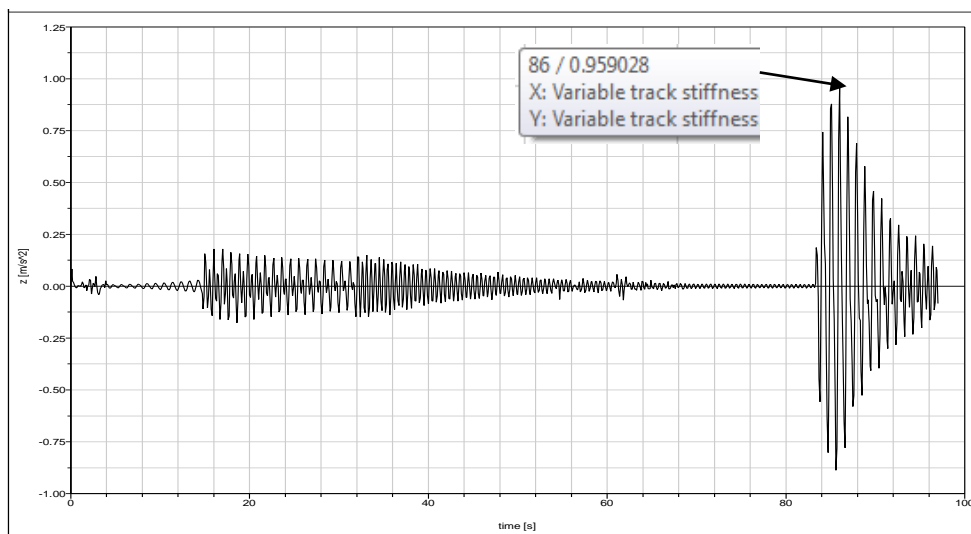


Figure 4.28 Vertical acceleration of the car body for variable track stiffness during EB

In the same way, before starting the motion the lateral acceleration of the car body was zero. Figure 4.29 and 4.30 show the lateral acceleration of the car body. For the case of constant track stiffness, the lateral acceleration is higher during the second acceleration however its value dies very soon. During the variable track stiffness case, the largest lateral acceleration of the car body was showed during the constant velocity motion of the vehicle.

The peak values for the two cases occurred at different locations i.e. at 20.7 second for constant track stiffness and 61.1 second for the variable track stiffness. Comparing the two outputs the variable track stiffness case has greater peak lateral acceleration value. In addition, if we compare the peak values during service and emergency braking case, the emergency braking case shows greater lateral acceleration of the car body.

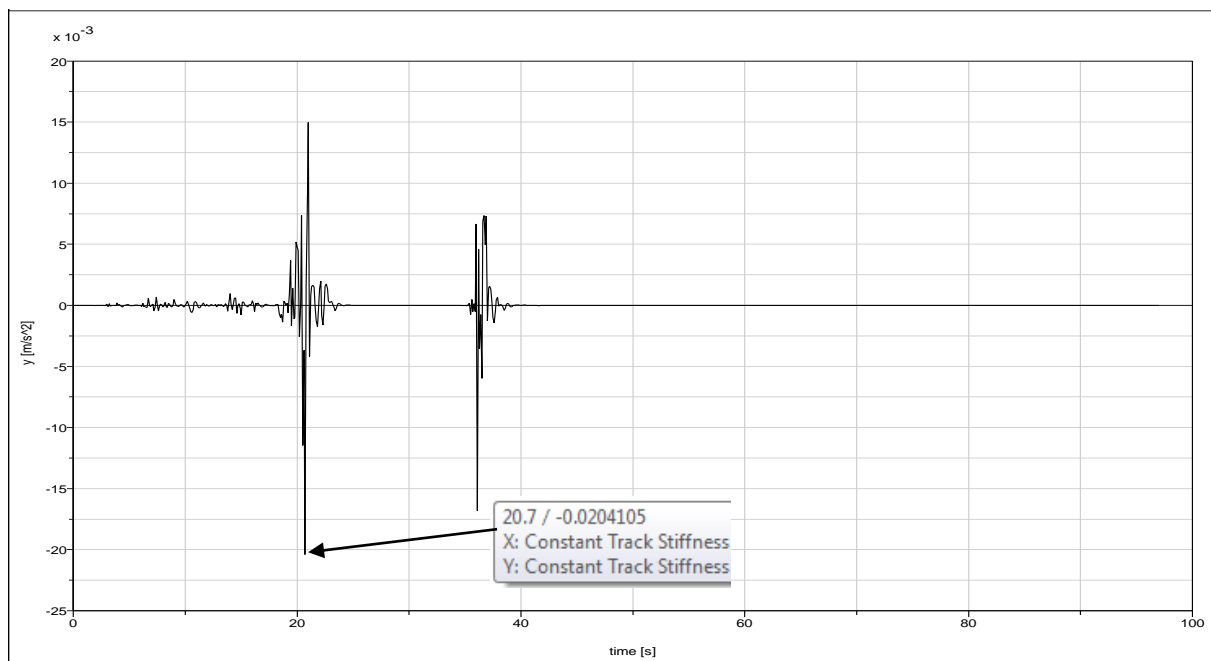


Figure 4.29 Lateral acceleration of the car body for the constant track stiffness during EB

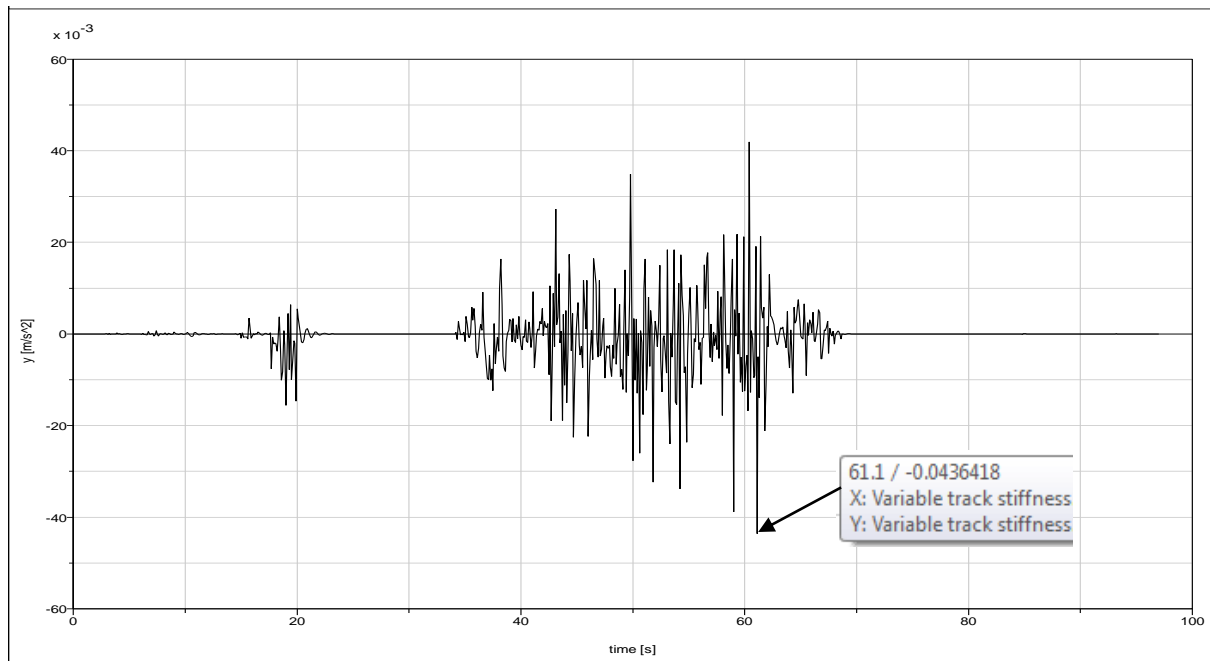


Figure 4.30 Lateral acceleration of the car body for the variable track stiffness during EB

4.2.2.3 Effect of track stiffness variation on bogie frame displacement

The bogie frame vertical displacement is shown in Figure 4.31 and 4.32. As it can be seen from the output, at static equilibrium the frame moved down for both cases. This is because of the preload calculation.

Again for both cases, the frame moved down upon vibrating during longitudinal acceleration and deceleration. An increment in the vertical displacement occurred during the change of longitudinal acceleration of the vehicle for both cases. After the increment however, the vertical displacement reduces continuously along the motion of the vehicle. The largest vertical frame displacement occurred around 84 second for both cases. As shown in the figures, the variable stiffness case has greater peak vertical displacement value. Again comparing the respective peak outputs of service and emergency braking cases, the emergency braking case showed larger value.

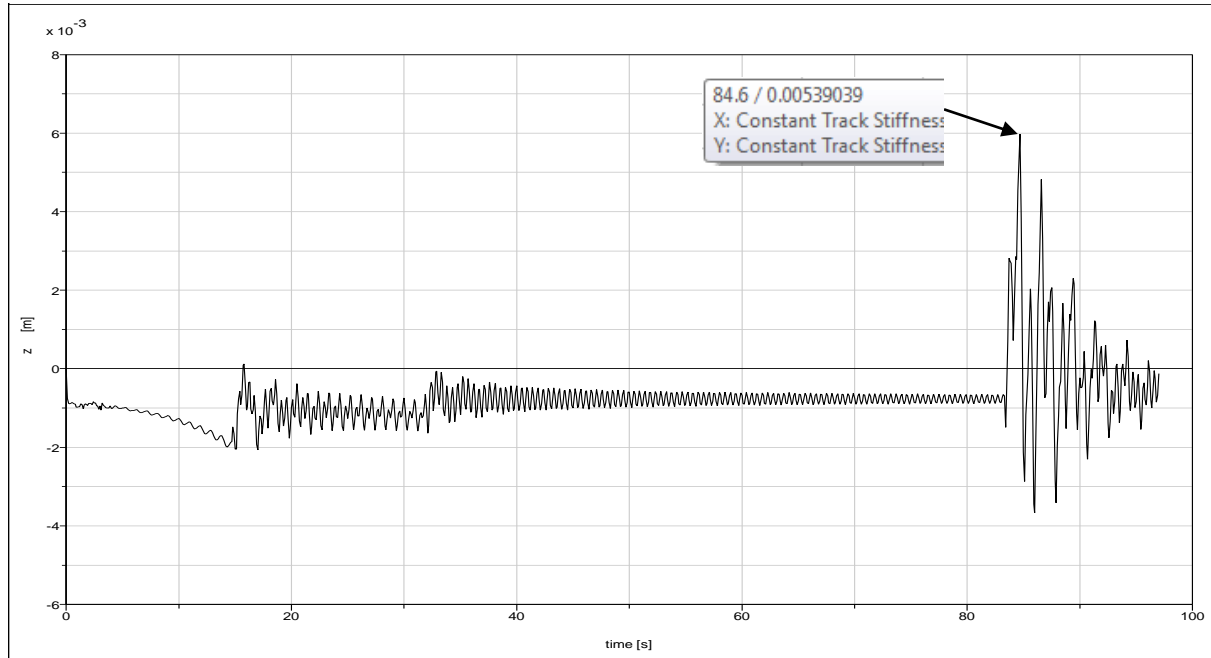


Figure 4.31 Vertical displacement of the frame for constant track stiffness during EB

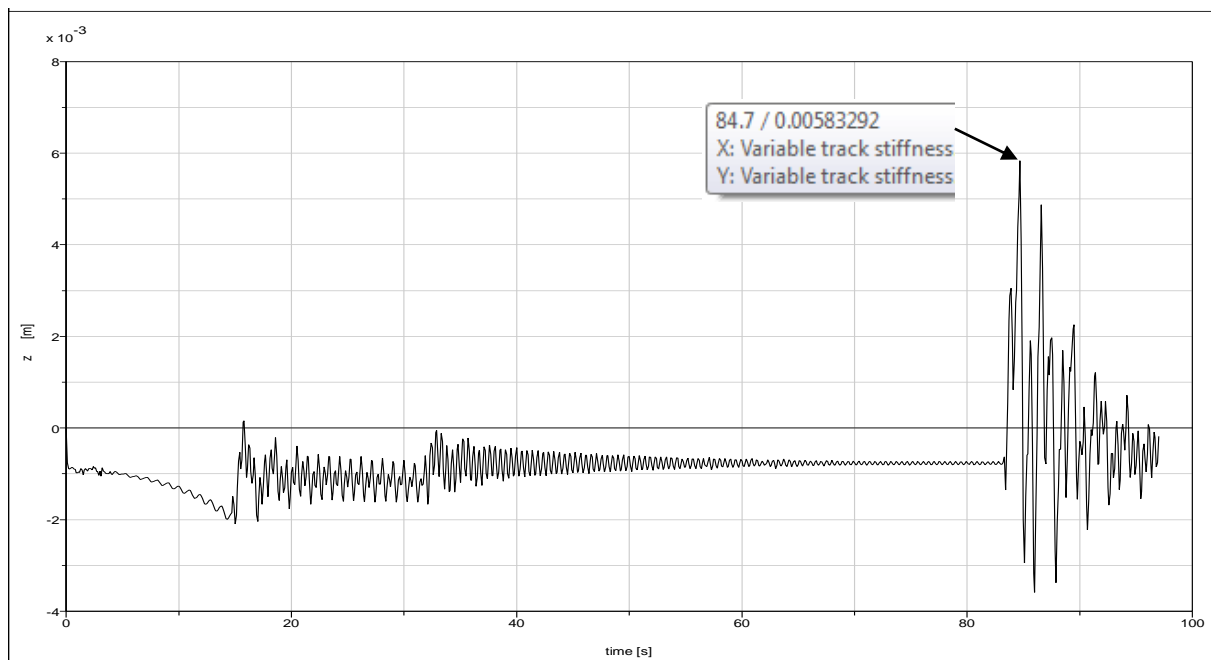


Figure 4.32 Vertical displacement of the frame for variable track stiffness during EB

Figure 4.33 and 4.34 shows the lateral displacement of the frame. The frame was at zero position before starting the motion for both cases. In the case of constant track stiffness, the frame lateral displacement was higher during the second acceleration and at the start of the constant velocity motion of the vehicle. For the variable track stiffness case, small lateral

displacement of the frame was shown at the second acceleration and it showed an erratic and higher lateral displacement when the vehicle moves at a constant velocity.

The peak values occurred at different locations for the constant and variable track stiffness cases, however the variable track stiffness case has greater value. In addition, comparing the respective outputs for the service and emergency braking cases, service braking constant stiffness case has showed larger value and emergency braking variable stiffness case showed larger value.

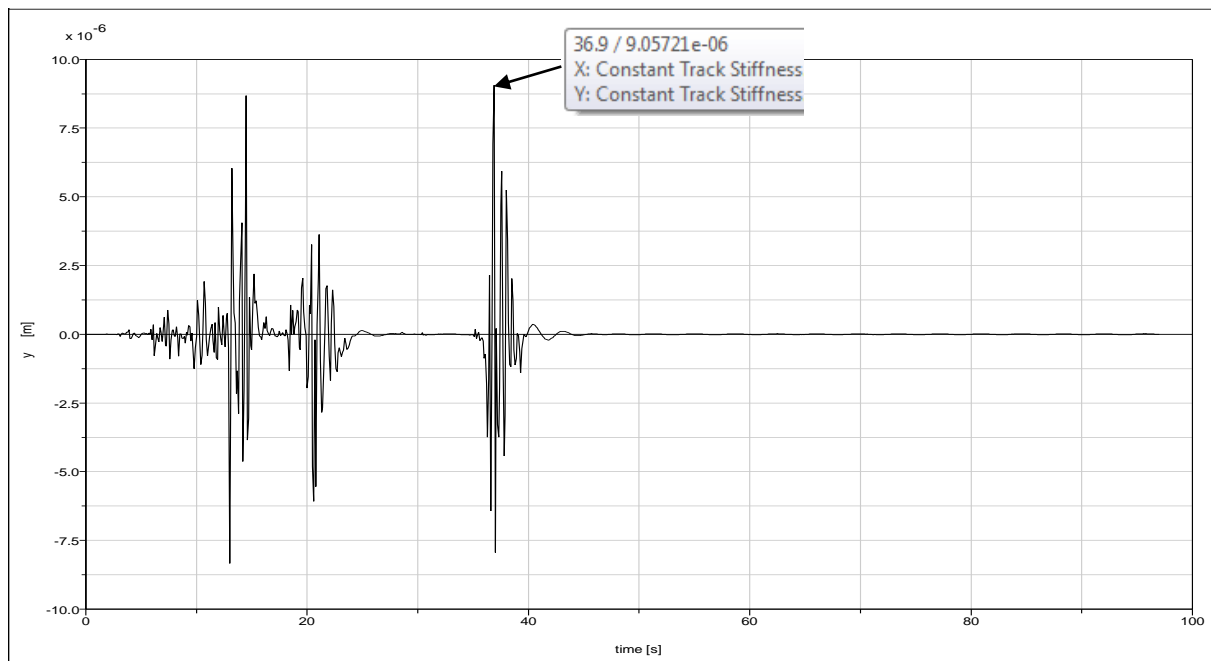


Figure 4.33 Lateral displacement of the frame for constant track stiffness during EB

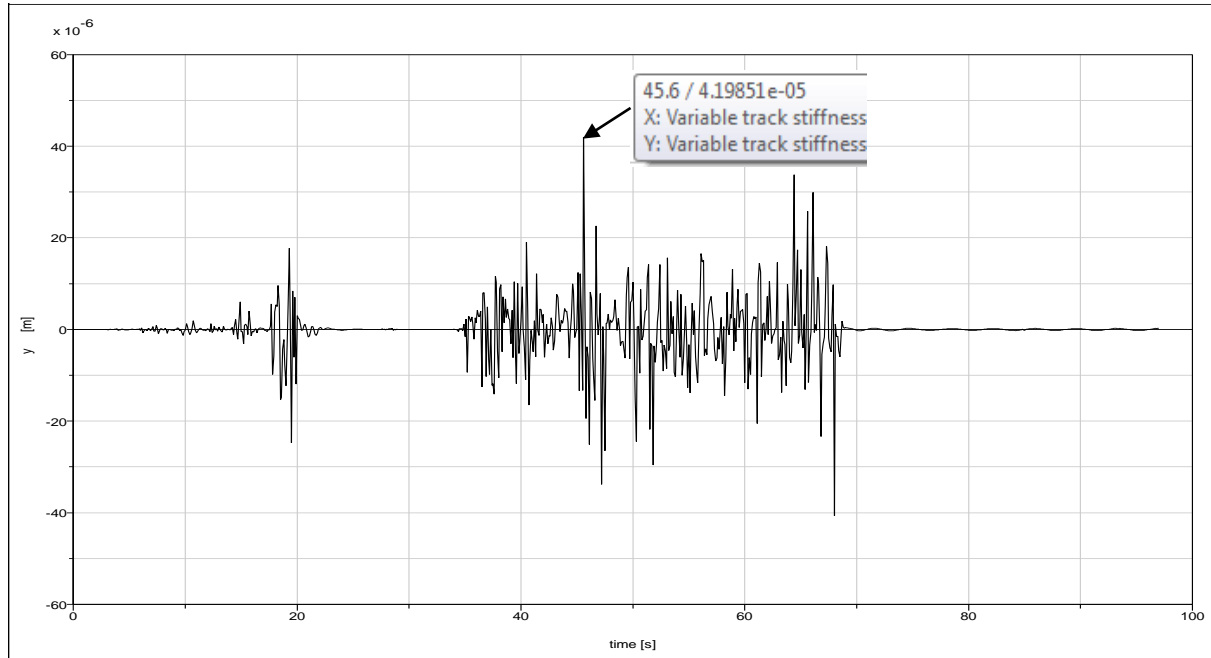


Figure 4.34 Lateral displacement of the frame for variable track stiffness during EB

4.2.2.4 Effect of track stiffness variation on bogie frame acceleration

As it can be seen from Figure 4.35 and 4.36, at the start of the motion the vertical acceleration of the frame for both cases is greater. This is because the vehicle starts motion. For both cases, higher vertical acceleration of the frame occurred at the second acceleration, beginning of constant velocity and when the vehicle is decelerating. The peak values occurred around 84 second for both cases, however the variable track stiffness case has greater value. During the constant velocity of the vehicle, the constant track stiffness case showed continuous decrease in the vertical acceleration and the variable track stiffness case showed an erratic vertical acceleration. Comparing the respective values for the service and the emergency cases, the emergency cases have showed greater peak value.

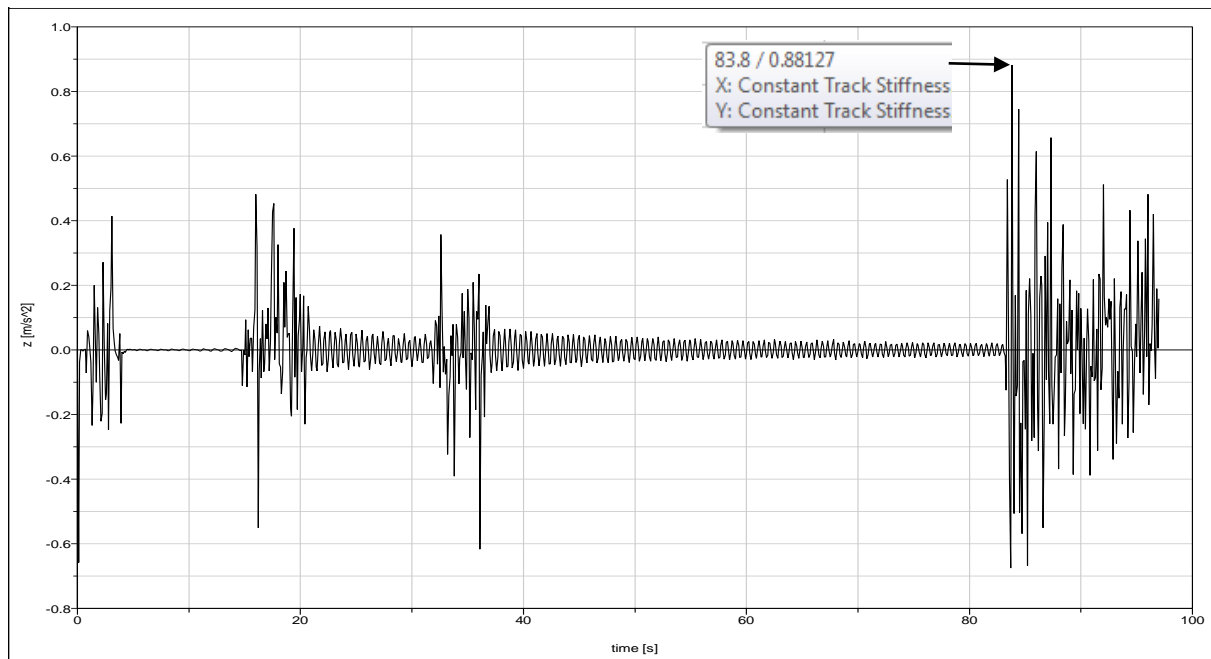


Figure 4.35 Vertical acceleration of the frame for constant track stiffness during EB

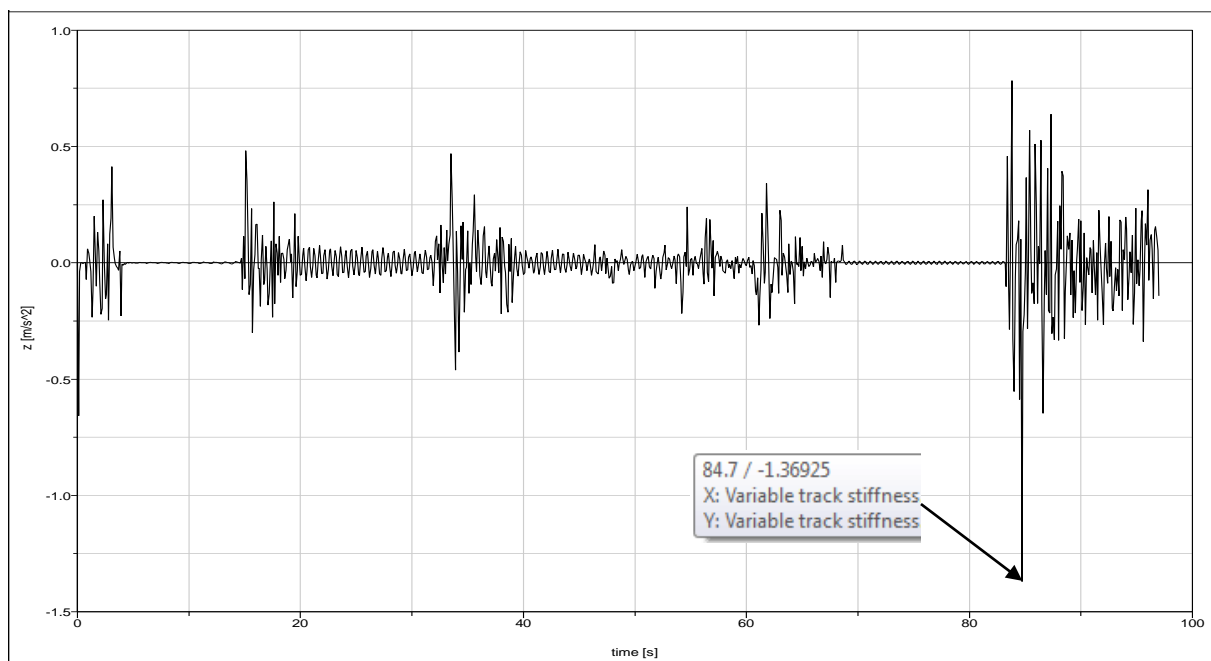


Figure 4.36 Vertical acceleration of the frame for variable track stiffness during EB

Figure 4.37 and 4.38 show the lateral acceleration of the frame. The constant track stiffness case shows the lateral acceleration of the frame is higher during the second acceleration and on the starting of the constant velocity motion of the vehicle and it remains zero for the rest. On the other hand, the variable track stiffness case shows an erratic lateral acceleration on the second

acceleration and on the constant velocity motion of the vehicle. Comparing the peak values of the two cases the variable track stiffness case has greater value. In addition, comparison of the respective outputs of the service and emergency braking peak values, the emergency value is greater.

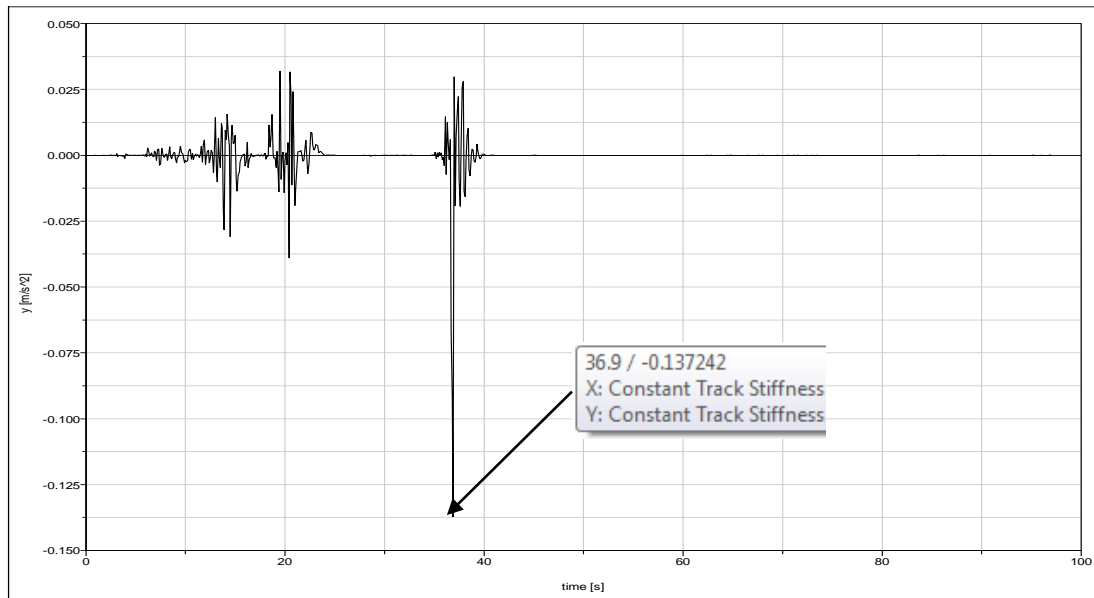


Figure 4.37 Lateral acceleration of the frame for constant track stiffness during EB

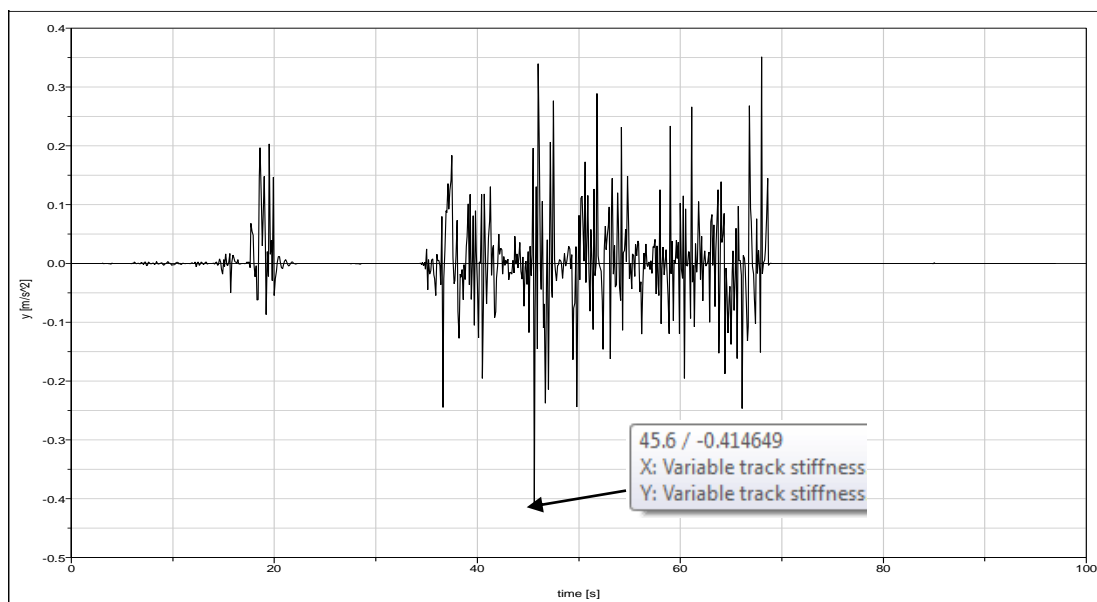


Figure 4.38 Lateral acceleration of the frame for variable track stiffness during EB

4.2.2.5 Summary of the outputs during emergency braking

As it can be seen from Table 4.3, the car body displacement and acceleration increased during the variable track stiffness. In addition, the bogie frame also showed increment both in acceleration and displacement during the variable track stiffness case.

Table 4.3 Summary of the output values for the constant and variable track stiffness during emergency braking motion.

Component	Measured quantity	Constant track stiffness	Variable track stiffness	% Increase or Decrease
Car body	Vertical displacement	0.0160519	0.018049	12.4
	Lateral displacement	0.0000545	0.0001834	236.5
	Vertical acceleration	0.951577	0.959028	0.8
	Lateral acceleration	0.0204105	0.0436418	113.8
Bogie frame	Vertical displacement	0.00539039	0.00583292	8.2
	Lateral displacement	9.05721×10^{-6}	4.19851×10^{-5}	363.6
	Vertical acceleration	0.88127	1.36925	55.4
	Lateral acceleration	0.137242	0.414649	202.1

4.2.3 Dynamic effect of track stiffness variation at increased longitudinal acceleration (IA)

In order to study how the variation of track stiffness affects the dynamic behavior of the vehicle at higher acceleration and deceleration, the values of the acceleration and deceleration taken during service braking analysis are doubled as shown in Figure 4.39.

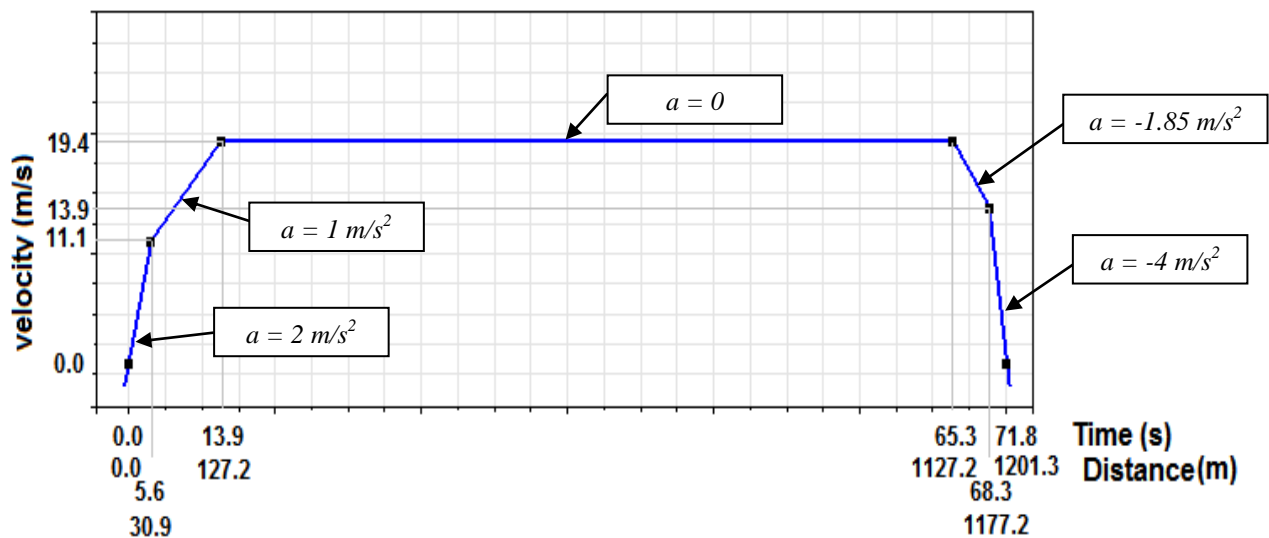


Figure 4.39 Velocity profile followed during an increased acceleration and deceleration

4.2.3.1 Effect of track stiffness variation on car body displacement

Figure 4.40 and 4.41 show the time histories of the displacement of the car body in the vertical direction for the constant and variable stiffness cases respectively. The largest displacement for both cases occurred during braking. However, comparing the values of the two cases, the variable track stiffness case has showed more vertical displacement. The two cases had showed similarity in the vertical displacement along the motion of the vehicle. When the respective outputs of the service braking and increased acceleration are compared, the increased acceleration case has greater peak value.

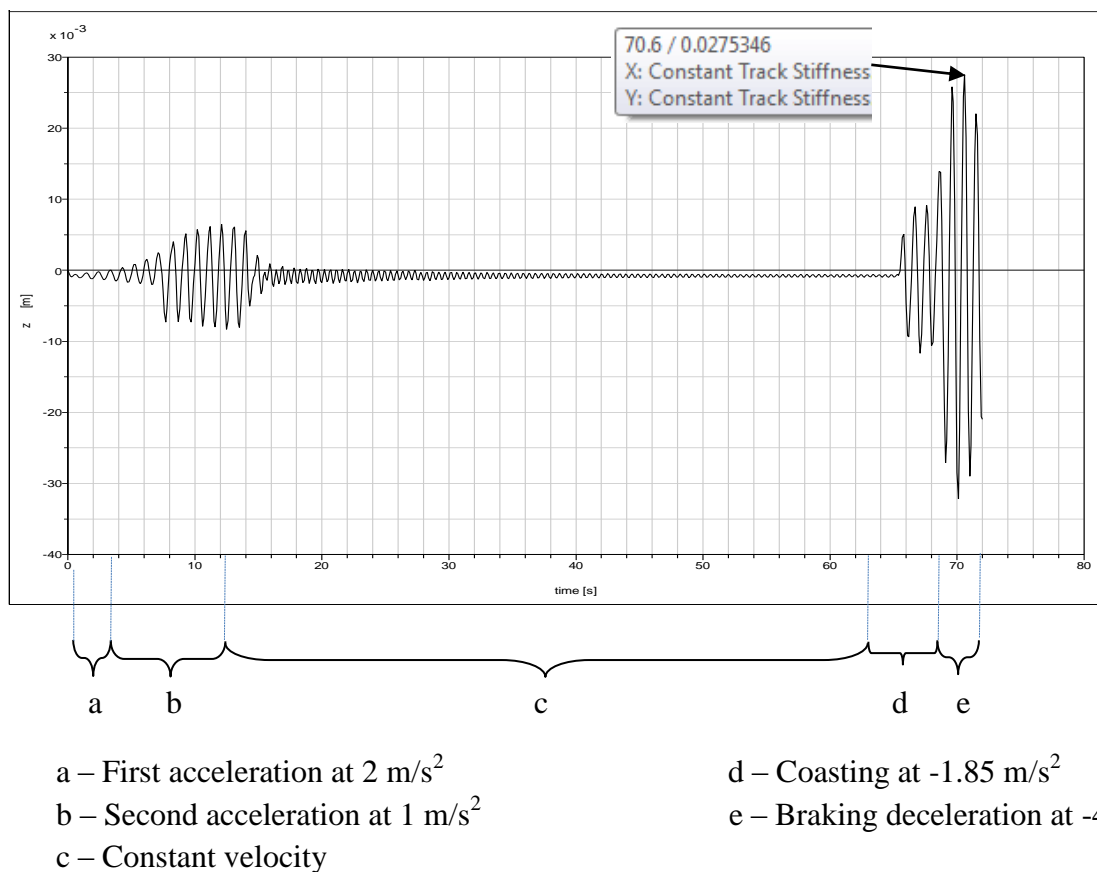


Figure 4.40 Vertical displacement of the car body for constant track stiffness during IA

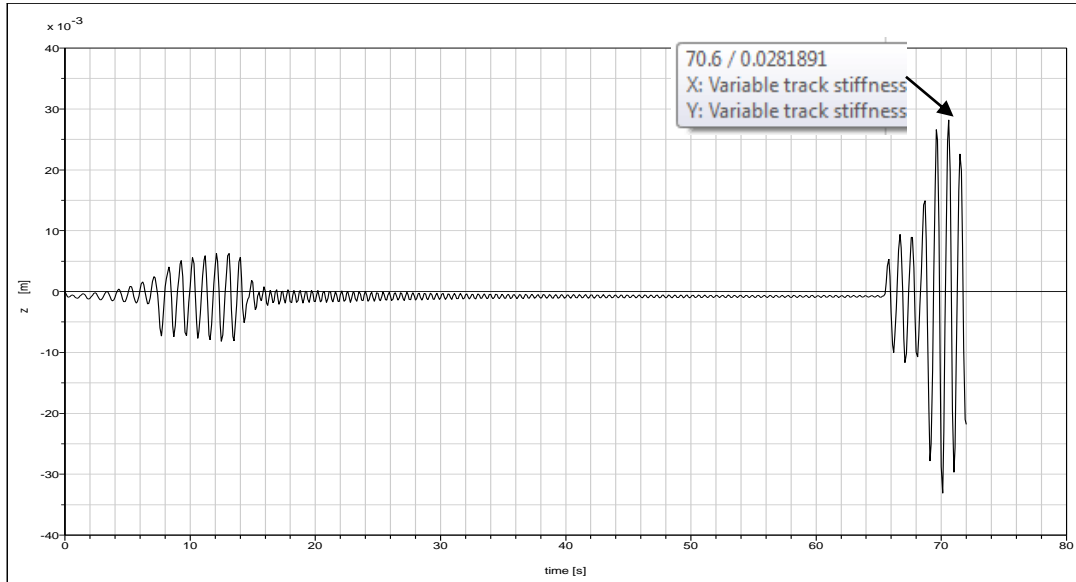


Figure 4.41 Vertical displacement of the car body for variable track stiffness during IA

Figure 4.42 and 4.43 show the lateral displacement of the car body for the constant track and variable track stiffness respectively. For both cases, lateral displacement of the car body is higher around 30 second. Comparing the two cases the variable track stiffness shows larger peak lateral acceleration value. In the same way, the increased acceleration case has greater lateral displacement peak value than the service braking case discussed above.

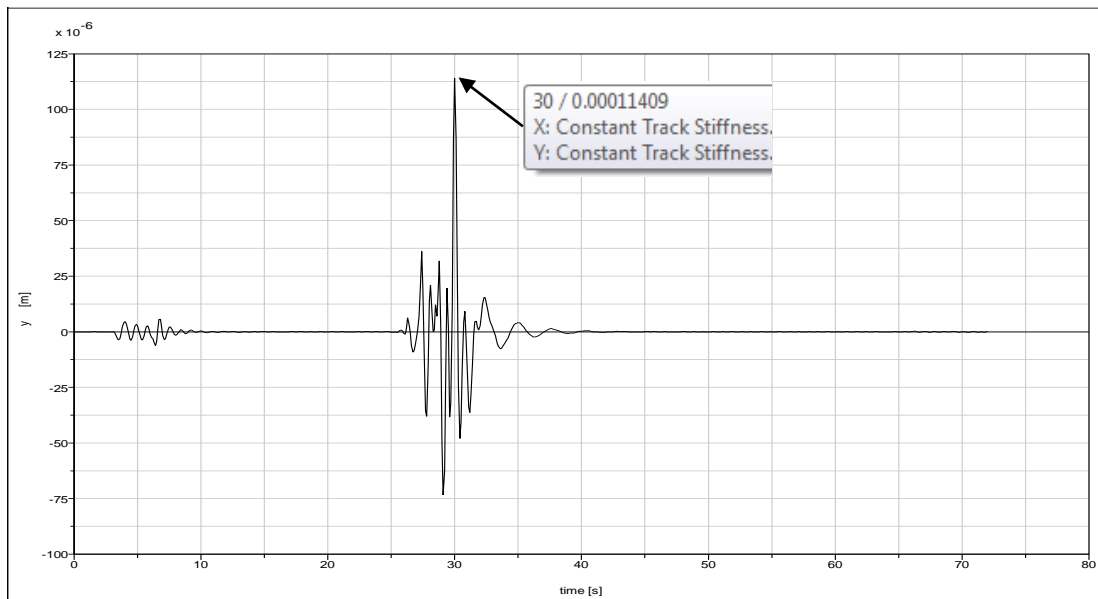


Figure 4.42 Lateral displacement of the car body for constant track stiffness during IA

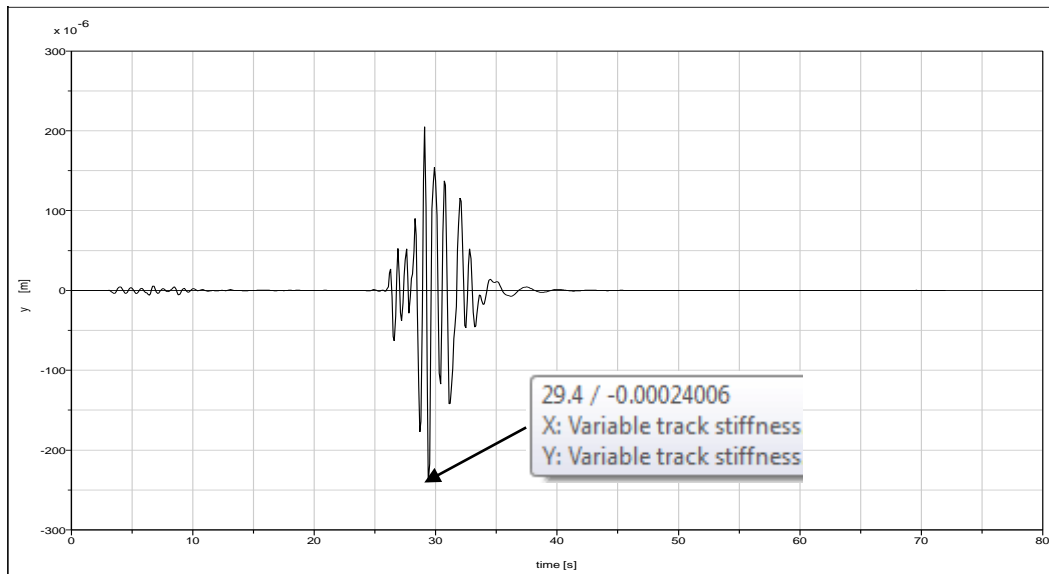


Figure 4.43 Lateral displacement of the car body for variable track stiffness during IA

4.2.3.2 Effect of track stiffness variation on car body acceleration

Figure 4.44 and 4.45 show car body vertical acceleration for the constant and variable case respectively. As it is shown in the figures, both cases showed largest vertical acceleration at 71 second, however the value for variable track stiffness is greater. Here also, the increased acceleration case has greater vertical acceleration peak value than the service braking case discussed above.

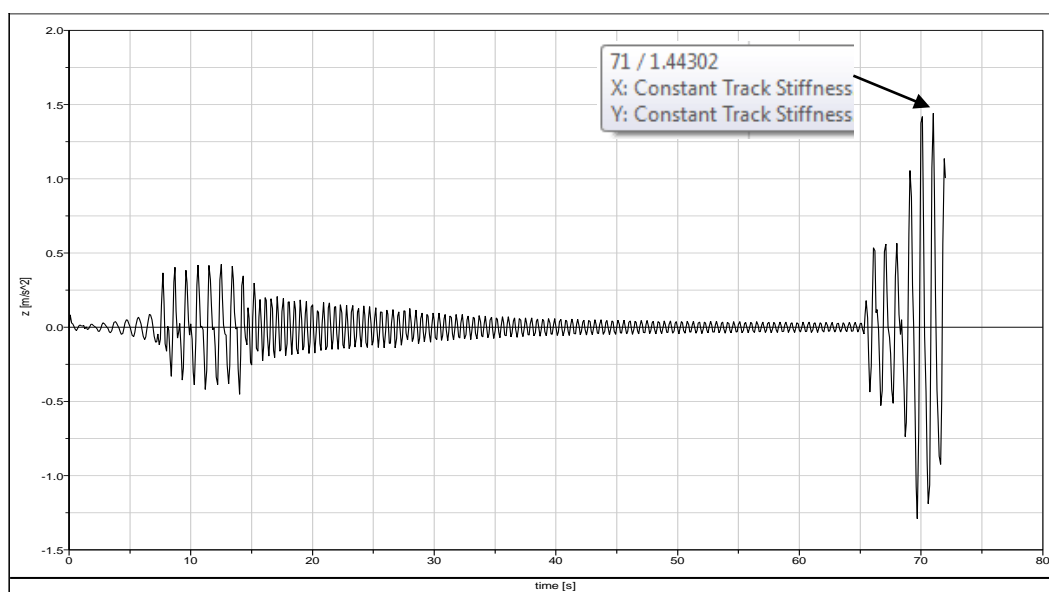


Figure 4.44 Vertical car body acceleration during constant track stiffness during IA

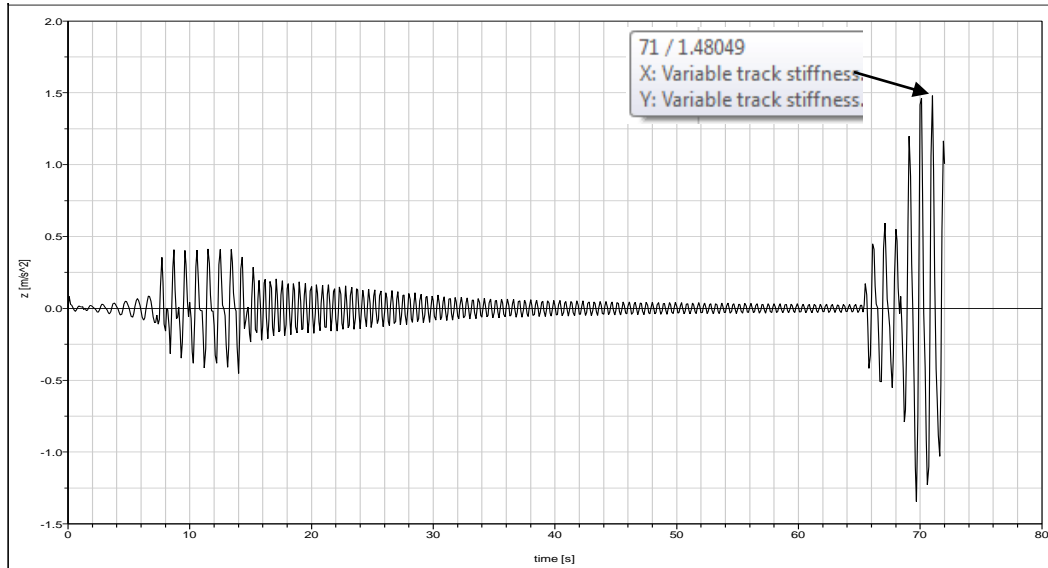


Figure 4.45 Vertical car body acceleration during variable track stiffness during IA

Figure 4.46 and 4.47 show the lateral acceleration of the car body. For both cases, the car body showed greater car body lateral acceleration at 29 second. Comparing the two outputs the variable track stiffness case has greater peak lateral acceleration value.

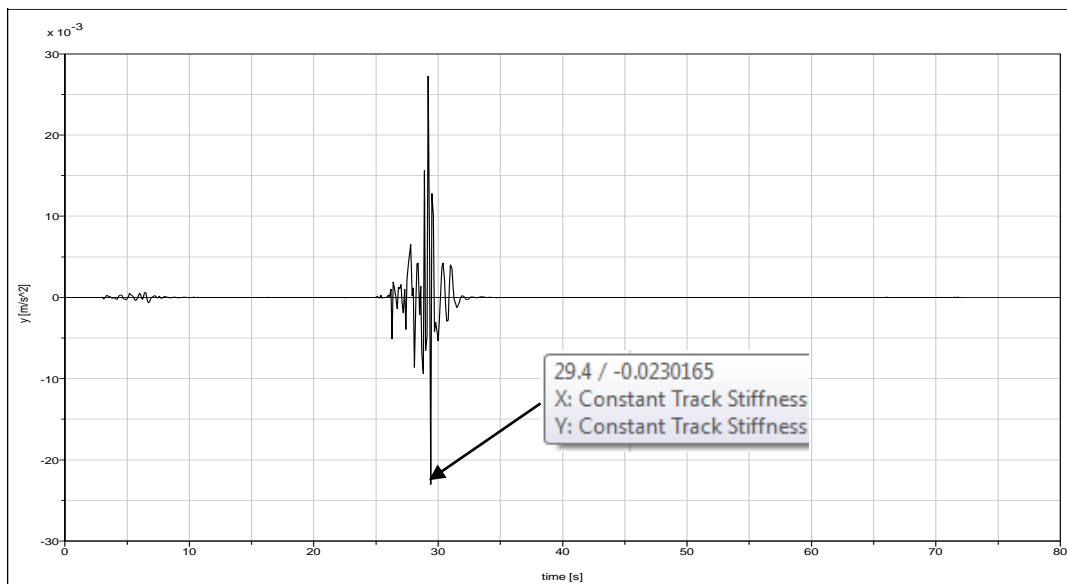


Figure 4.46 Lateral car body acceleration during constant track stiffness during IA

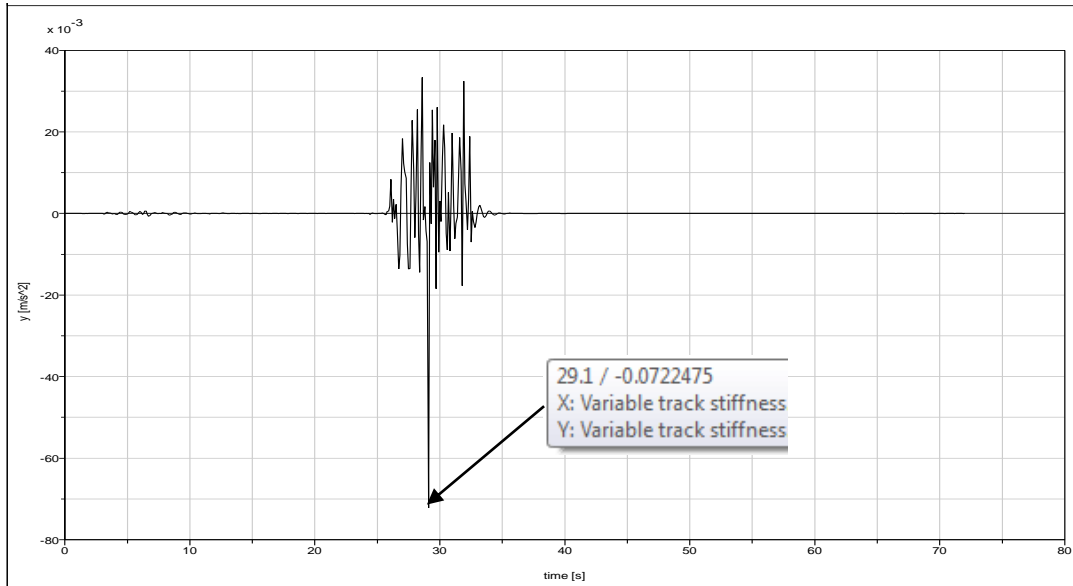


Figure 4.47 Lateral car body acceleration during variable track stiffness during IA

4.2.3.3 Effect of track stiffness variation on bogie frame displacement

The bogie frame vertical displacement is shown in Figure 4.48 and 4.49. For both cases, the peak values occurred around 69 second, however the variable track stiffness case has greater value.

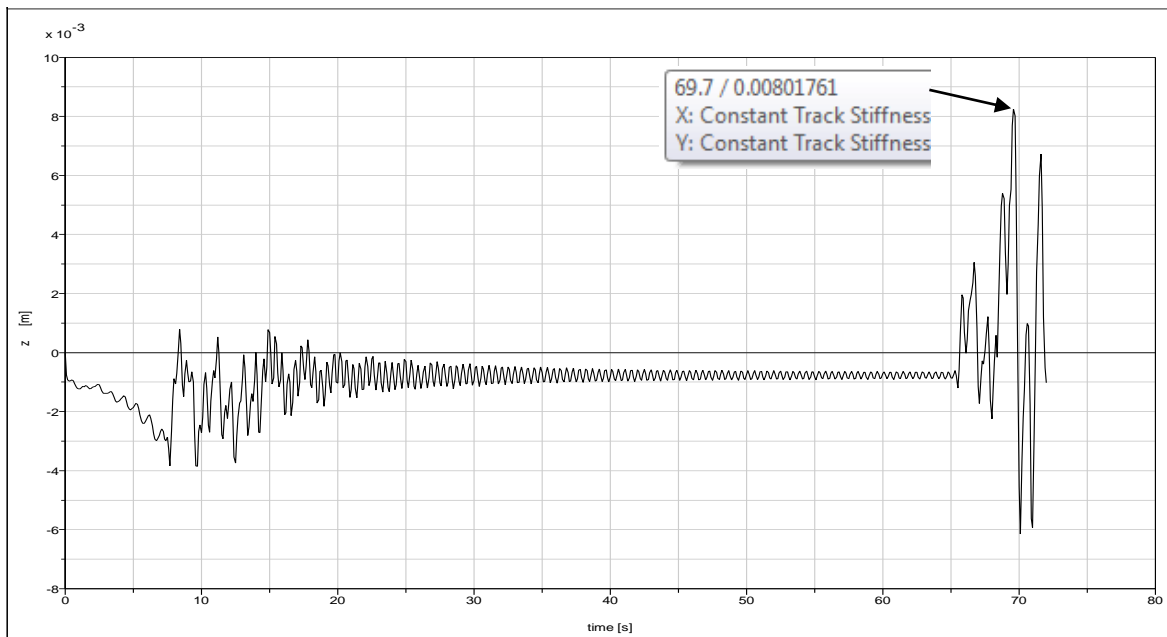


Figure 4.48 Vertical frame displacement during constant track stiffness during IA

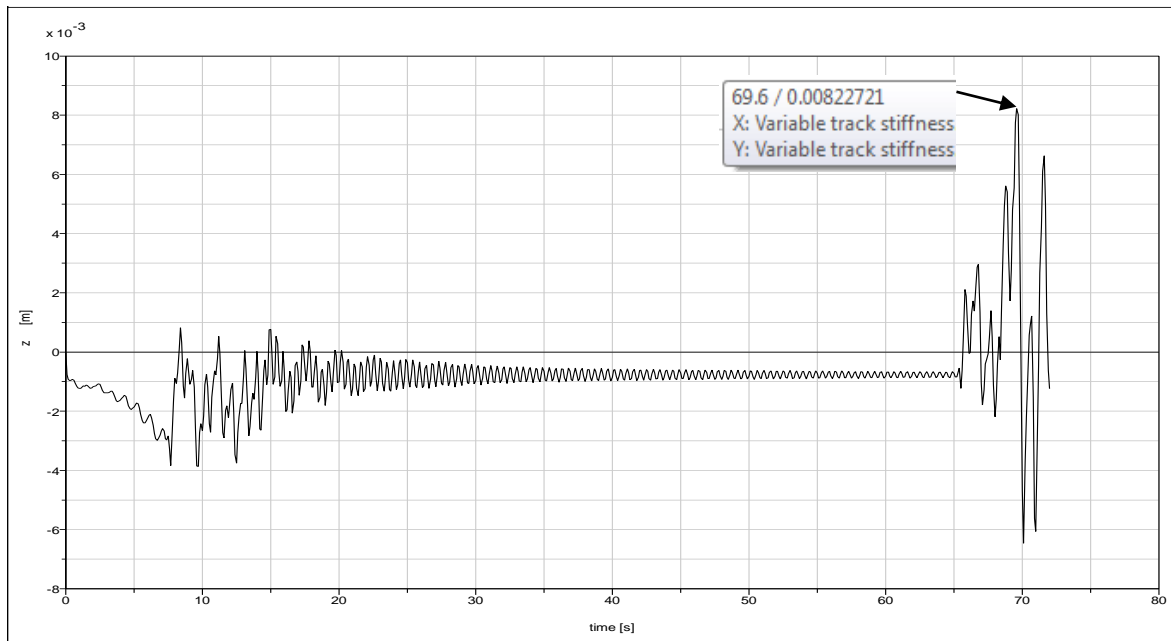


Figure 4.49 Vertical frame displacement during variable track stiffness during IA

In the lateral direction, the frame was at zero position before starting the motion. In both cases, the peak lateral displacement occurred 29 second. Comparing the two cases the constant track stiffness case shows greater value. Figure 4.50 and 4.51 show the lateral displacement of the frame.

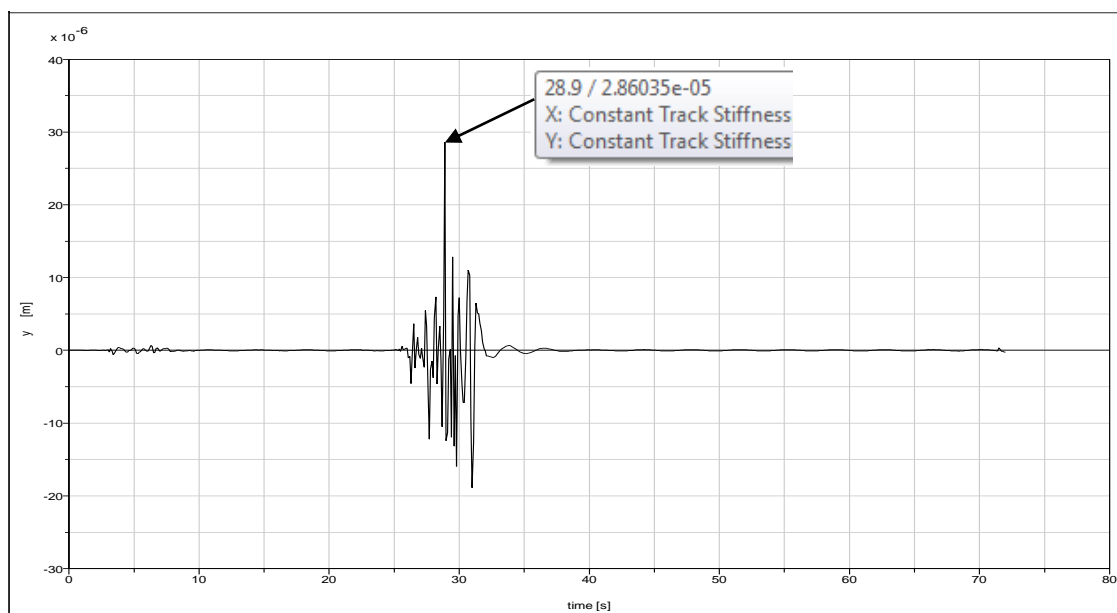


Figure 4.50 Lateral displacement of the frame during constant track stiffness during IA

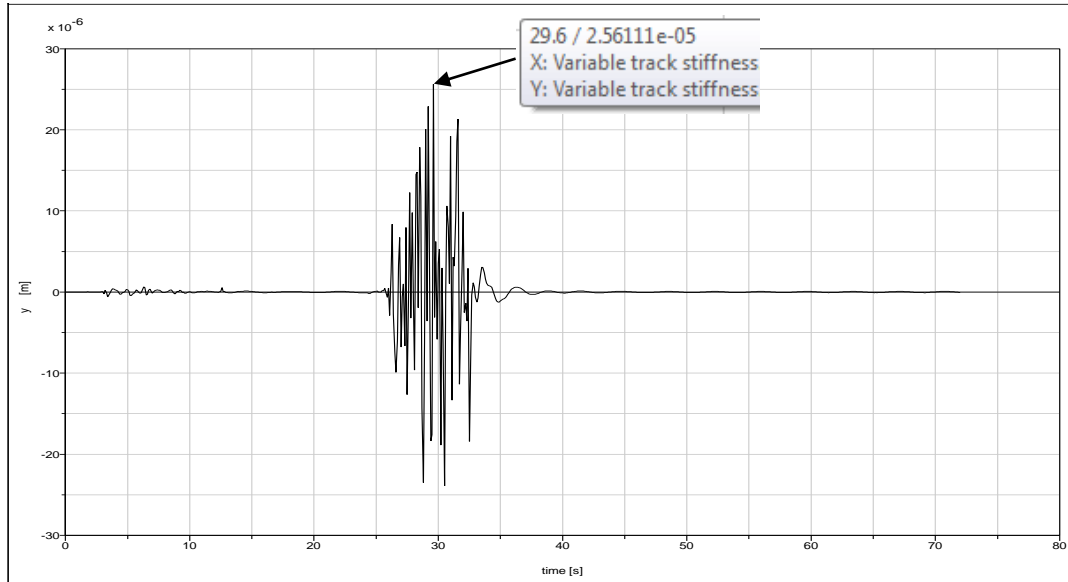


Figure 4.51 Lateral displacement of the frame during variable track stiffness during IA

4.2.3.4 Effect of track stiffness variation on bogie frame acceleration

Figure 4.52 and 4.53 shows the vertical acceleration of the frame. At the start of the motion the vertical acceleration of the frame for both cases is greater. For both cases, the vertical acceleration is greater during acceleration, coasting and deceleration. If we take the peak of the two cases, the variable stiffness case has greater value.

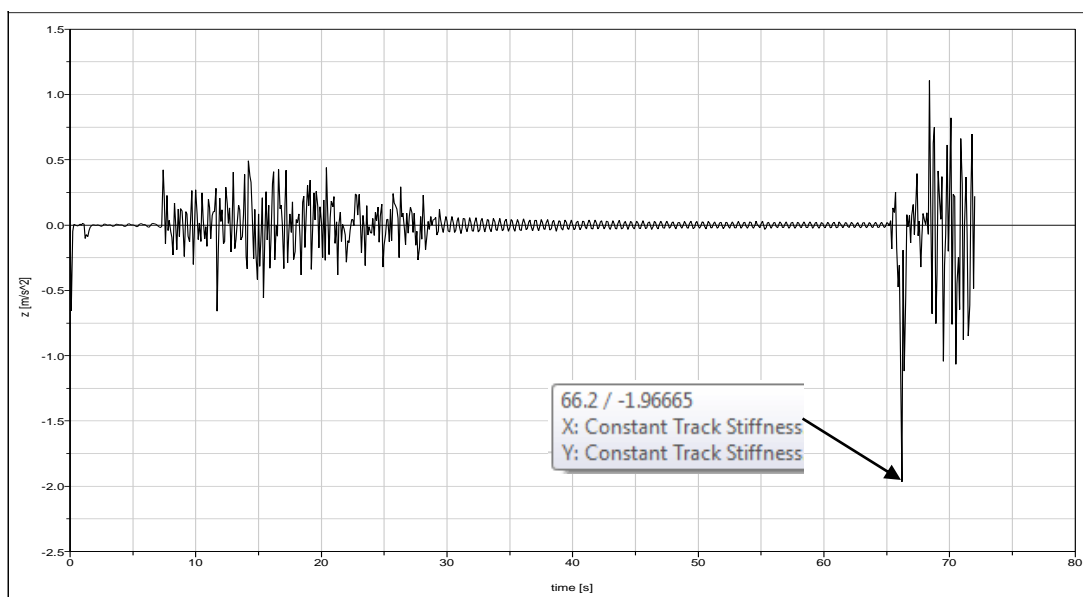


Figure 4.52 Vertical acceleration of the frame during constant track stiffness during IA

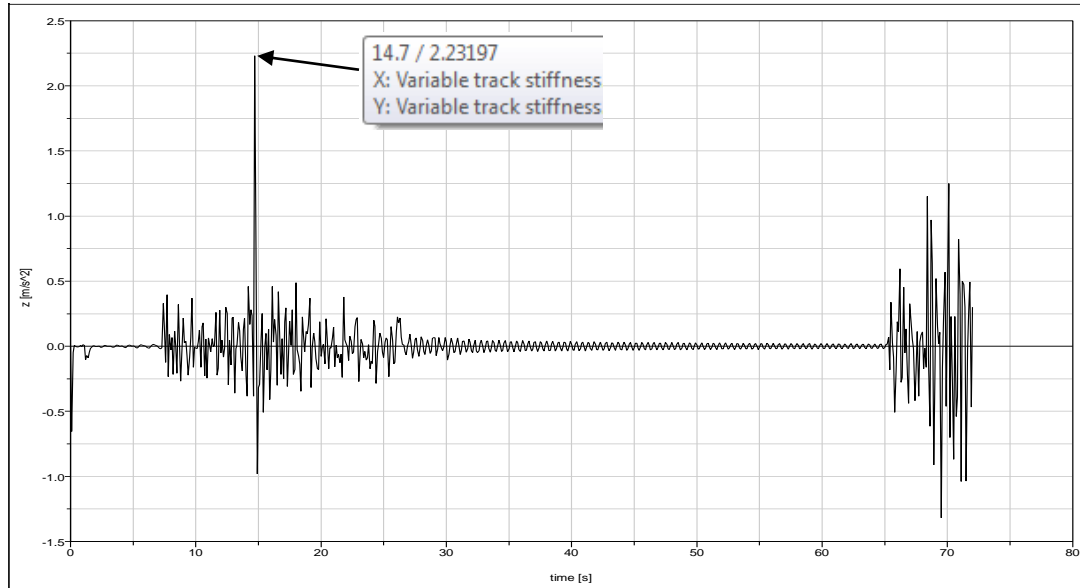


Figure 4.53 Vertical acceleration of the frame during variable track stiffness during IA

As shown in Figure 4.54 and 4.55 the greatest lateral acceleration for both cases occurred around 29 second. Comparing the peaks of the two cases the variable track stiffness case has greater value.

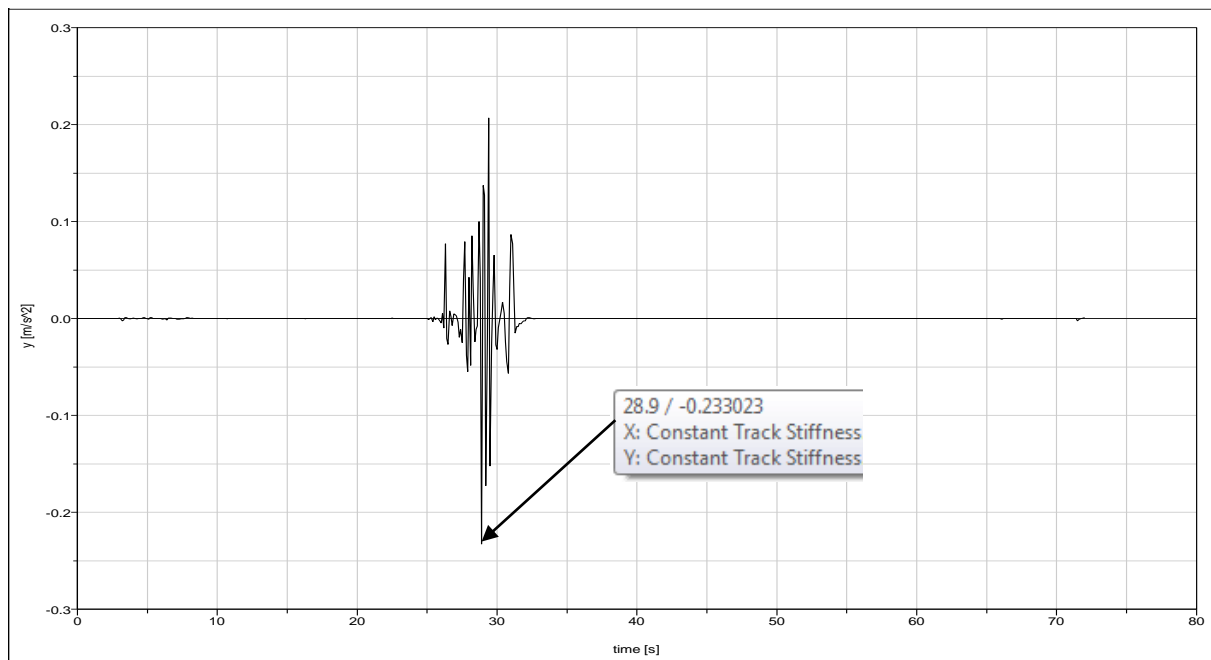


Figure 4.54 Lateral acceleration of the frame during constant track stiffness during IA

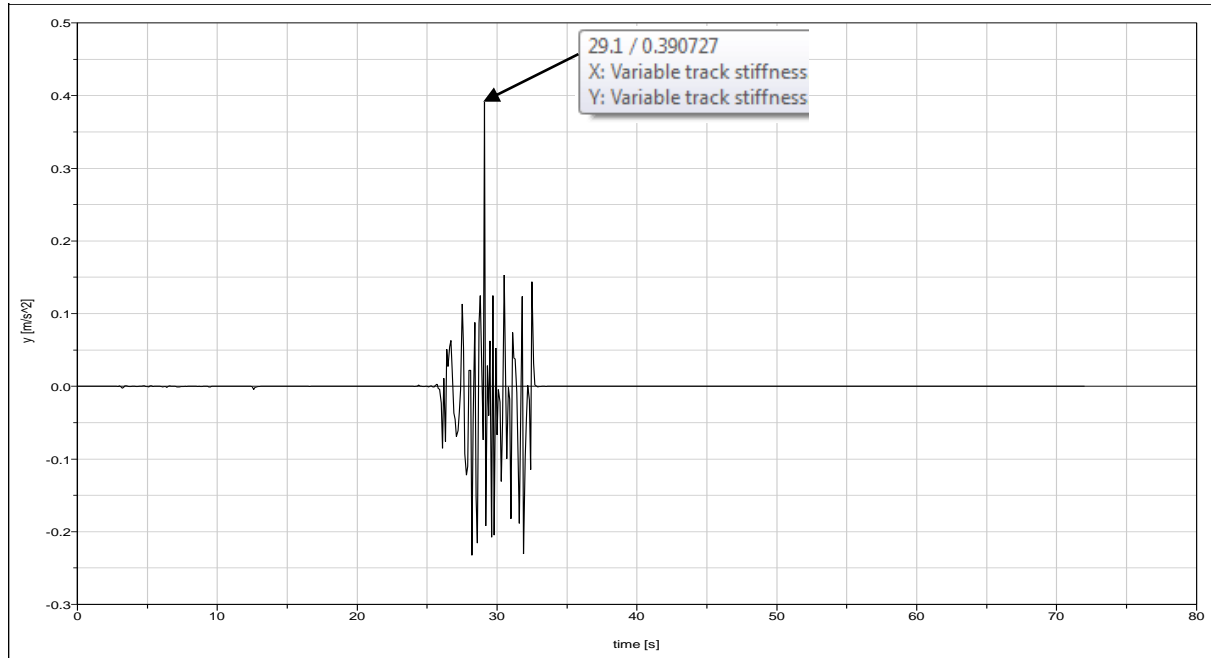


Figure 4.55 Lateral acceleration of the frame during variable track stiffness during IA

4.2.3.5 Summary of the outputs during increased acceleration

As it can be seen from Table 4.4, the car body displacement and acceleration increased during the variable track stiffness. In addition, the bogie frame also showed increment both in acceleration and displacement during the variable track stiffness case except for the lateral displacement.

Table 4.4 Summary of the output values for the constant and variable track stiffness during increased acceleration.

Component	Measured quantity	Constant track stiffness	Variable track stiffness	% Increase or Decrease
Car body	Vertical displacement	0.0275346	0.0281891	2.4
	Lateral displacement	0.00011409	0.00024006	110.4
	Vertical acceleration	1.44302	1.48049	2.6
	Lateral acceleration	0.0230165	0.0722475	213.9
Bogie frame	Vertical displacement	0.00801761	0.00822721	2.6
	Lateral displacement	2.86035×10^{-5}	2.56111×10^{-5}	-10.5
	Vertical acceleration	1.96665	2.23197	13.5
	Lateral acceleration	0.233023	0.390727	67.7

4.2.3.6 Comparison of outputs during service braking and increased acceleration

As it can be seen from Table 4.5 and 4.6, the car body displacement and acceleration increased during increasing the longitudinal acceleration of the vehicle. The outputs for the bogie frame shows increase in some quantities and decrease in the others.

Table 4.5 Comparison of the outputs for variable track stiffness during service braking and increased longitudinal acceleration.

Component	Measured quantity	Variable track stiffness		% Increase or Decrease
		Service braking case	Increased acceleration	
Car body	Vertical displacement	0.00337141	0.0281891	736.1
	Lateral displacement	0.00012715	0.00024006	88.8
	Vertical acceleration	0.262728	1.48049	463.5
	Lateral acceleration	0.0364441	0.0722475	98.2
Bogie frame	Vertical displacement	0.00146325	0.00822721	462.3
	Lateral displacement	0.000026	2.56111×10^{-5}	-1.5
	Vertical acceleration	0.538076	2.23197	314.8
	Lateral acceleration	0.252796	0.390727	54.6

Table 4.6 Comparison of the outputs for constant track stiffness during service braking and increased longitudinal acceleration.

Component	Measured quantity	Constant track stiffness		% Increase or Decrease
		Service braking case	Increased acceleration	
Car body	Vertical displacement	0.00283561	0.0275346	871.0
	Lateral displacement	0.00009838	0.00011409	16.0
	Vertical acceleration	0.2414	1.44302	497.8
	Lateral acceleration	0.0186788	0.0230165	23.2
Bogie frame	Vertical displacement	0.0012221	0.00801761	556.1
	Lateral displacement	0.0000375	2.86035×10^{-5}	-23.7
	Vertical acceleration	0.573527	1.96665	242.9
	Lateral acceleration	0.274375	0.233023	-15.1

CHAPTER FIVE

CONCLUSION, RECOMMENDATION AND FUTURE WORK

5.1 Conclusion

The objective of this research was to analyze the effect of track stiffness variation on the vehicle dynamic conditions. This was achieved by developing two multi-body dynamic models using SIMPACK software. The track stiffness of one of the models was made constant and the other was made variable. Both of the models allowed to move with the same velocity profile on a straight track. In addition, the wheels and rails taken for simulation are perfect i.e. without defect. The model for the variable track stiffness case was validated for the plausibility of the output. The models reported in this paper are capable of predicting the dynamic responses of different components of the vehicle at different locations. From the output results the following conclusions can be made:-

- ❖ The car body displacement in the vertical and lateral directions varies during acceleration, constant velocity motion and deceleration. Highest vertical displacements occurred after the vehicle changes longitudinal acceleration. The car body displacements in the vertical and lateral direction are greater for the variable track stiffness case than the constant track stiffness. The output results show that the variation of track stiffness increases the vertical as well as the lateral displacement of the car body.
- ❖ The variation of car body acceleration also shows variation during positive, zero and negative longitudinal acceleration of the vehicle. Greater change of vertical acceleration of the vehicle occurred during the longitudinal acceleration change of the vehicle based on the velocity profile. For both vertical and lateral motion of the vehicle an increase of acceleration was shown for variable track stiffness case than the constant track stiffness. From these observations, it can be concluded that the variability of track stiffness increases both lateral and vertical acceleration of the car body.
- ❖ Comparing the values of displacement and acceleration outputs vertically and laterally for the bogie frame no uniformity were shown i.e. some outputs show the variable stiffness case has greater value and in some cases lesser value. From the outputs it can be concluded that, track stiffness variation may not have effect on the bogie frame.

- ❖ Comparison of effect of track stiffness variation on different velocity profile shows a velocity profile having greater longitudinal acceleration values have greater vertical and lateral car body displacement and acceleration. However, the motion of the frame showed non uniform output. From these observations we can conclude that, increasing acceleration longitudinally increases the vertical and longitudinal displacement and acceleration of the car body and may not have effect on the bogie frame.

Generally, we can conclude that track stiffness variation increases the vertical and lateral displacement and acceleration of the car body.

5.2 Recommendation

There is no doubt that the findings of this research can be used to provide valuable information to the design and maintenance engineer of railway vehicle as well as track. From the results of the research the following recommendations can be stated:-

- ❖ If stiffness is evaluated for a track, it could be used as a parameter to optimize the maintenance activities both for the track and the vehicle. As it was discussed, the variation of track stiffness and damping creates greater vibration on the vehicle components. This vibration even will be more if there are local weaknesses and hanging sleepers along the track. So, maintenance division of railway companies should give priority to retain a constant stiffness along the track. Stiffness measurements should be performed regularly and used as a tool when maintenance is planned. By keeping a constant track stiffness, there will be less damage to the track and the vehicle and the passengers will experience more comfortable journey.
- ❖ Proper care has to be taken during the design and laying of the track. Tracks should have to be designed to have less stiffness variability. The vehicle components also have to be designed considering the dynamic loading caused by the track stiffness variation.

5.3 Future work

Track stiffness is a complex area and understandings about track stiffness and its effect on track performance are far from complete. This is observed by the fact that there is not yet a European standard for vertical track stiffness available. [23] In this research, SIMPACK model was used to show the effect of track stiffness variation on the dynamic behavior of the vehicle. Researches that can be carried out considering track stiffness variation are numerous. Some of them are listed below:-

- ❖ In this research only the displacements and accelerations of the car body was analyzed. However, the model can be modified to get car body deflections and stresses caused by track stiffness variation. This can be made by adding flexibility to the car body. A more detailed model including for instance seats and passengers could also be interesting.
- ❖ Achieving good ride comfort is an important goal in the design of new competitive rail vehicles. In this research, it was showed that track stiffness variation increases the vertical as well as the lateral acceleration of the car body. This has negative effect on the passengers comfort. So, this research can be extended to analyze the ride index and passenger comfort level caused by track stiffness variation and design the suspension systems in order to optimize the passenger comfort.
- ❖ In this research, only track stiffness variation caused by sleeper spacing was considered. However, variation of track stiffness occurs on other places such as switches, crossings and transition places. So, the effect of stiffness variation on these places can be studied. In addition to this, unsupported sleepers cause great variation of stiffness, which is the cause of increased track damage and high stresses in some of the vehicle component. Several unsupported sleepers in a region violate the safety requirements. This could be potential research area.

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Appendix

A. The governing equation of motion for the mathematical model developed in section 3.3

For the car body

Taking force equilibrium in the vertical direction

$$M_c \ddot{Z}_c - K_s(Z_c - Z_{FFR}) - K_s(Z_c - Z_{FFL}) - K_s(Z_c - Z_{BFR}) - K_s(Z_c - Z_{BFL}) + M_c g = 0$$

Rearranging we have

$$M_c \ddot{Z}_c - 4K_s Z_c + K_s(Z_{FFR} + Z_{FFL} + Z_{BFR} + Z_{BFL}) + M_c g = 0 \quad \text{A.1}$$

Taking moment about x-axis

$$I_{cx} \ddot{\theta}_c - 4K_s a \tan \theta \times a - 4K_s Z_c a + K_s(Z_{FFR} + Z_{FFL} + Z_{BFR} + Z_{BFL})a - 4\rho_s \dot{\theta}_c a \tan \theta \times a = 0$$

For small angle θ , $\tan \theta = \theta$, so

$$I_{cx} \ddot{\theta}_c - 4K_s a^2 \theta - 4K_s Z_c a + K_s(Z_{FFR} + Z_{FFL} + Z_{BFR} + Z_{BFL})a - 4\rho_s \dot{\theta}_c a R W^2 = 0 \quad \text{A.2}$$

Similarly taking moment about y-axis

$$I_{cy} \ddot{\alpha}_c - 4K_s l^2 \alpha - 4K_s Z_c l + K_s(Z_{FFR} + Z_{FFL} + Z_{BFR} + Z_{BFL})l - 4\rho_s \dot{\alpha}_c l^2 = 0 \quad \text{A.3}$$

For the front right bogie frame

Taking force equilibrium in the vertical direction

$$\begin{aligned} M_f \ddot{Z}_{FFR} - \rho_p(\dot{Z}_{FFR} - \dot{Z}_{FW1}) - \rho_p(\dot{Z}_{FFR} - \dot{Z}_{FW2}) + K_s(Z_c - Z_{FFR}) \\ - K_p(Z_{FFR} - Z_{FW1}) - K_p(Z_{FFR} - Z_{FW2}) + 2K_s l \alpha + M_f g = 0 \end{aligned}$$

Rearranging we have,

$$\begin{aligned} M_f \ddot{Z}_{FFR} - 2\rho_p \dot{Z}_{FFR} + \rho_p(\dot{Z}_{FW1} + \dot{Z}_{FW2}) + K_s(Z_c - Z_{FFR}) \\ - 2K_p Z_{FFR} + K_p(Z_{FW1} + Z_{FW2}) + 2K_s l \alpha + M_f g = 0 \end{aligned} \quad \text{A.4}$$

Taking moment about y-axis

$$I_f \ddot{\alpha}_{FFR} - 2K_p j \tan \alpha_{FFR} \times j - 2\rho_p j^2 \dot{\alpha}_{FFR} = 0$$

Rearranging we have,

$$I_f \ddot{\alpha}_{FFR} - 2K_p j^2 \alpha_{FFR} - 2\rho_p j^2 \dot{\alpha}_{FFR} = 0 \quad \text{A.5}$$

Similarly for the front left bogie frame

Taking force equilibrium in the vertical direction

$$\begin{aligned} M_f \ddot{Z}_{FFL} - 2\rho_p \dot{Z}_{FFL} + \rho_p (\dot{Z}_{FW1} + \dot{Z}_{FW2}) + K_s (Z_c - Z_{FFL}) \\ - 2K_p Z_{FFL} + K_p (Z_{FW1} + Z_{FW2}) + 2K_s l \alpha + M_f g = 0 \end{aligned} \quad A.6$$

Taking moment about y-axis

$$I_f \ddot{\alpha}_{FFL} - 2K_p j^2 \alpha_{FFL} - 2\rho_p j^2 \dot{\alpha}_{FFL} = 0 \quad A.7$$

Similarly for the rear right bogie frame

Taking force equilibrium in the vertical direction

$$\begin{aligned} M_f \ddot{Z}_{RFR} - 2\rho_p \dot{Z}_{RFR} + \rho_p (\dot{Z}_{RW1} + \dot{Z}_{RW2}) + K_s (Z_c - Z_{RFR}) \\ - 2K_p Z_{RFR} + K_p (Z_{RW1} + Z_{RW2}) + 2K_s l \alpha + M_f g = 0 \end{aligned} \quad A.8$$

Taking moment about y-axis

$$I_f \ddot{\alpha}_{RFR} - 2K_p j^2 \alpha_{RFR} - 2\rho_p j^2 \dot{\alpha}_{RFR} = 0 \quad A.9$$

Similarly for the rear left bogie frame

Taking force equilibrium in the vertical direction

$$\begin{aligned} M_f \ddot{Z}_{RFL} - 2\rho_p \dot{Z}_{RFL} + \rho_p (\dot{Z}_{RW1} + \dot{Z}_{RW2}) + K_s (Z_c - Z_{RFL}) \\ - 2K_p Z_{RFL} + K_p (Z_{RW1} + Z_{RW2}) + 2K_s l \alpha + M_f g = 0 \end{aligned} \quad A.10$$

Taking moment about y-axis

$$I_f \ddot{\alpha}_{RFL} - 2K_p j^2 \alpha_{RFL} - 2\rho_p j^2 \dot{\alpha}_{RFL} = 0 \quad A.11$$

For the front bogie first wheelset

$$\begin{aligned} M_w \ddot{Z}_{FW1} - 2\rho_p \dot{Z}_{FW1} - 2\rho_t \dot{Z}_{FW1} - K_p (Z_{FW1} - Z_{FFR}) \\ - K_p (Z_{FW1} - Z_{FFL}) - K_{TFL1} Z_{FW1} - K_{TFR1} Z_{FW1} + M_w g = 0 \end{aligned}$$

Rearranging we have,

$$\begin{aligned} M_w \ddot{Z}_{FW1} - 2(\rho_p + \rho_t) \dot{Z}_{FW1} - 2K_p Z_{FW1} \\ + K_p (Z_{FFR} + Z_{FFL}) - (K_{TFL1} + K_{TFR1}) Z_{FW1} + M_w g = 0 \end{aligned} \quad A.12$$

Taking moment about x-axis through the center of mass of the wheelset

$$I_w \ddot{w}_{FW1} - 2\rho_t t^2 \dot{w}_{FW1} - 2\rho_p b^2 \dot{w}_{FW1} - 2K_p b^2 w_{FW1} - K_{TFR1} t^2 w_{FW1} - K_{TFL1} t^2 w_{FW1}$$

$$-K_{TFR1}tW_{FW1} - K_{TFL1}tW_{FW1} = 0 \quad \text{A.13}$$

Similarly, for the front bogie second wheelset

$$\begin{aligned} &M_w \ddot{Z}_{FW2} - 2(\rho_p + \rho_t) \dot{Z}_{FW2} - 2K_p Z_{FW2} \\ &+ K_p (Z_{FFR} + Z_{FFL}) - (K_{TFR2} + K_{TFL2}) Z_{FW2} + M_w g = 0 \end{aligned} \quad \text{A.14}$$

Taking moment about x-axis through the canter of mass of the wheelset

$$\begin{aligned} &I_w \ddot{W}_{FW2} - 2\rho_t t^2 \dot{W}_{FW2} - 2\rho_p b^2 \dot{W}_{FW2} - 2K_p b^2 W_{FW2} - K_{TFR2} t^2 W_{FW2} - K_{TFL2} t^2 W_{FW2} \\ &- K_{TFR2} t W_{FW2} - K_{TFL2} t W_{FW2} = 0 \end{aligned} \quad \text{A.15}$$

Similarly, for the rear bogie first wheelset

$$\begin{aligned} &M_w \ddot{Z}_{RW1} - 2(\rho_p + \rho_t) \dot{Z}_{RW1} - 2K_p Z_{RW1} \\ &+ K_p (Z_{RRR} + Z_{RRL}) - (K_{TRL1} + K_{TFL1}) Z_{RW1} + M_w g = 0 \end{aligned} \quad \text{A.16}$$

Taking moment about x-axis through the canter of mass of the wheelset

$$\begin{aligned} &I_w \ddot{W}_{RW1} - 2\rho_t t^2 \dot{W}_{RW1} - 2\rho_p b^2 \dot{W}_{RW1} - 2K_p b^2 W_{RW1} - K_{TRR1} t^2 W_{RW1} - K_{TRL1} t^2 W_{RW1} \\ &- K_{TRR1} t W_{RW1} - K_{TRL1} t W_{RW1} = 0 \end{aligned} \quad \text{A.17}$$

Similarly, for the rear bogie second wheelset

$$\begin{aligned} &M_w \ddot{Z}_{RW2} - 2(\rho_p + \rho_t) \dot{Z}_{RW2} - 2K_p Z_{RW2} \\ &+ K_p (Z_{RRR} + Z_{RRL}) - (K_{TRL2} + K_{TFL2}) Z_{RW2} + M_w g = 0 \end{aligned} \quad \text{A.18}$$

Taking moment about x-axis through the canter of mass of the wheelset

$$\begin{aligned} &I_w \ddot{W}_{RW2} - 2\rho_t t^2 \dot{W}_{RW2} - 2\rho_p b^2 \dot{W}_{RW2} - 2K_p b^2 W_{RW2} - K_{TRR2} t^2 W_{RW2} - K_{TRL2} t^2 W_{RW2} \\ &- K_{TRR2} t W_{RW2} - K_{TRL2} t W_{RW2} = 0 \end{aligned} \quad \text{A.19}$$

B. SIMPACK GUI used to insert the values in section 3.4.4

